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Why Shoot Mosquitoes?

IF a man were shooting in the heart of a fine game country it seems as though he would be foolish to start in to shoot mosquitoes and overlook the birds and other game.

A bill emanating from the Department of Commerce has been introduced in the House of Representatives requiring the numbering of small, undocumented vessels propelled by machinery. To properly enforce such a bill and its sister bill requiring the inspection of hulls and machinery and the licensing of the operating engineers on motor boats carrying twenty or more passengers would cost at least a million dollars (£205,000) a year.

Why shoot off such an immense amount of ammunition on such comparative trivialities when there is such urgent —we might safely say criminal—need for bringing up to date the laws pertaining to the inspection of steam vessels? If Congress plans to spend any money along these lines, let it be on shooting the big game. Let us have an inspection service such as the Inspector-General has been asking for for years. When this has been accomplished Congress can, in a fit of wanton extravagance, if it sees fit, put up the million dollars or more to let the Inspector-General shoot mosquitoes.

Ship Stability

ONE outcome of the recent *Eastland* disaster has been the introduction of a bill in Congress providing for the establishment of a board of naval architects in the office of the Supervising Inspector General of the United States Steamboat Inspection Service for the purpose of passing upon the plans of self-propelled vessels of 100 tons and over to be built in the United States for operation under the American flag. One of the main reasons for the establishing of this board is undoubtedly the expectation that such a board will serve to prevent another occurrence like the *Eastland* disaster, which was caused by the instability of the vessel.

That a thorough investigation of the question of the stability of ships is greatly needed is shown in the article on ship stability published elsewhere in this issue. As the author points out, although the expression "safe stability" is frequently used, it would be difficult to define it in exact terms. Neither the metacentric height, nor the angle through which a ship would roll without losing her righting moment, nor the maximum righting lever, would be of any value as a criterion of safe stability when taken separately. Even the knowledge of all three of these qualities is not sufficient to form an opinion as to a vessel's safety. Once the vessel is in service various circumstances affecting her loading and ballasting may alter her behavior within wide limits under various conditions of service and numerous examples are on record where experienced captains have been deceived as to the condition of their vessels with disastrous results. Theoretical conditions alone cannot determine the safe stability of a vessel, and it is only by combining theory and practice that a workable standard can be determined that will serve as a practical safeguard for all types of vessels and for all conditions of service.

To establish such a standard for safe stability would require an enormous amount of investigation of ships in commission, and it seems highly improbable that the proposed board of naval architects for the United States Steamboat Inspection Service would have the facilities or qualifications for carrying out such investigations or for doing more than is being done at present by competent classification societies. Such a project was begun in Germany just before the outbreak of the war and it was intended to establish a central bureau for collecting information on the stability of ships at sea, in addition to which calculations would be made so that the whole problem of stability might be placed on a proper basis. Such an investigation would be of inestimable value if carried out by the best qualified body of experienced and trained men obtainable, but the provisions of the proposed bill now being considered in Congress do not appear likely to attract men of such qualifications to this service.

Naval Experimental Laboratory

WHILE the recommendation made by the newly formed Naval Consulting Board for the establishment of a five-million-dollar research laboratory at tidewater for naval experimental work may seem to the general public like a radical innovation, as a matter of fact a laboratory of this kind, costing over half a million dollars and known as the Naval Engineering Experiment Station has been in operation at Annapolis for nearly eight years. The only reason why the work of this laboratory has not been conducted on the scale proposed by Mr. Edison and his associates for the new research laboratory has been the lack of appropriations for the work. If the sums recommended by the Naval Consulting Board for this work are provided by Congress, what better facilities could be offered than a further development of the present experiment station?

The experiment station is located on the Severn River, opposite the sea front of the Naval Academy. The grounds contain about ten acres and have a water frontage of about 1,600 feet. At present the laboratory is housed in a single building 316 feet long by 66 feet wide and an addition called the boiler house 66 feet by 34 feet and a boiler house extension 86 feet by 66 feet, with a 26-foot lean-to on three sides. The main building contains the offices, laboratories, storeroom, tool room, machine shop, wood-working shop, etc. The boiler house contains

a Niclausse boiler, a Babcock & Wilcox boiler, feed pumps, air compressors, condensers and a dynamo room with a 75 kilowatt turbo generator. The boiler house extension contains two airtight fireroom compartments, in one of which is installed an oil-burning Mosher boiler. The lean-tos around the boiler house contain a forge shop, an instrument storage and calibration room, a foundry, sheet metal shop and special testing devices. Some idea of the work done in this laboratory may be gained from the following: gage glasses have been thoroughly investigated, with the result that a type of gage glass employing mica to insulate the glass from the action of steam has been found which will give good service and a minimum of breakdown and attention. Packings and lubricating oils have been thoroughly investigated, with the result that the values of different types are better known and considerable savings have been made in the purchase of supplies of such materials. Corrosion of boiler materials has received a good deal of attention, both as to the materials themselves and as to the treatment of boiler waters, with the result that there has been developed a boiler compound which, when properly used, will prevent corrosion. This is known as the Navy Standard Compound, and is now being recognized and applied to engineering work in civil life. Photomicrography has been applied extensively in determining the proper causes of failures of metals. Tests have been made as to the relative efficiencies of coal stored under salt water. Boiler feed pumps, turbo blowers, water and oil metering devices, fuel oil burners, reducing valves, safety valves and various types of ordinary valves have been thoroughly investigated.

All tests are conducted from the service point of view and under conditions as nearly approaching conditions on board ship as the facilities of the experiment station will permit. Although the station has had little time for research work, the investigations into the merits of the appliances tested involved discussions leading to conclusions and recommendations which are worked out with a view to the probable improvements of the appliances. It is difficult to see how any other place that could be recommended would offer facilities so well suited to the purposes of a research laboratory as the site of the present experiment station. It is at tide-water within 500 feet of a 30-foot dredged channel and a waterway could easily be developed for the accommodation of battleships. It is located on one of the most strongly protected bays in the country, with a large, clear stretch of water for maneuvers, unhampered by shipping. It is adjacent to a large city and is within an hour's ride of the headquarters of the Navy Department. By developing the present facilities the research laboratory would be available at once.

Steamship Terminal Improvement

COMPETENT authorities agree that a saving of from 10 to 20 cents a ton can be saved in handling miscellaneous freight at marine terminals by the proper use of mechanical appliances. As government statistics show that over 300,000,000 tons of miscellaneous freight are handled at marine terminals in the United States every year, the possibilities of effecting great economies by the adoption of suitable mechanical appliances at the terminals are evident.

It is reasonable to assume that at least one-sixth of the total freight handled at marine terminals each day must be taken away from the terminals by street vehicles to destinations widely scattered within a radius of several miles. A case is cited elsewhere in this issue in which a steamship company increased the territory for collecting freight for transportation by its ships fourfold by using a motor truck service for the collecting of freight. In this case the commodity shipped was fruit which was grown on the shores of Lake Michigan. By adopting the motor truck service, fruit growers located twenty-five to thirty miles away from the steamship terminal were enabled to ship their products to Chicago economically by boat. The expense of this collection and delivery service was shared by both the fruit growers and the steamship company, and both benefited by it so greatly that it is probable that the service which was started as an experiment will be continued and extended in other places.

A similar case, cited in the same article, is that of a steamship line which operates freight steamers between Philadelphia and Wilmington on the Delaware River. This company maintains a collection and delivery service and recently has added motor trucks to its equipment for this purpose, thereby greatly increasing the economy and usefulness of the service. The practice of having the transportation companies handle the service of collecting and delivering the freight has long been established abroad, and has been tried in this country by at least one large railroad company, where it proved highly successful, but was discontinued because the Interstate Commerce Commission would not permit the absorption of the charges in the freight tariffs and a separate charge was found impracticable. There seems to be no reason, however, why some such arrangement could not be adopted with profit by certain of the steamship companies, and such a plan has been outlined in some detail by the secretary of the Electric Vehicle Association of America.

This association has given special consideration to the problem of eliminating congestion and the great waste associated with the handling of freight at marine and railroad terminals. The secretary of the Association goes so far as to predict that the time is coming when existing freight terminals will be eliminated, and instead of the present method of transfer of freight by miscellaneous concerns and individuals, the transportation companies themselves will deliver the freight direct to the consignee. It is expected that such an arrangement would work a great many reforms. In the first place, the loss of from $90,000,000 (£18,450,000) to $100,000,000 (£20,500,000) annually, chargeable to the delay in serving trucks at railway and marine terminals, would be eliminated by the operation of a fleet of electric stevedores on a scientific schedule, making relatively uniform demand for freight for the entire twenty-four hours. In cases where difficulty is experienced in obtaining additional pier accommodations, storage yards and warehouses, as is the case in many of our largest seaports, notably New York, a greater volume of freight could be handled in a less area than now utilized at the terminals, if the freight upon arrival at the terminals were properly handled and delivered. The adoption of electric stevedores would assure the delivery of the freight much more quickly than is the case with the existing system. Also night deliveries from the steamship terminals to the large establishments in the business districts of congested cities would greatly reduce the delays, as the freight could be delivered and the shipments handled when the streets of the city were not congested.

If the freight were to be distributed in the proposed scientific manner, the economies that would be effected in minimizing traffic losses, it is claimed, would be almost beyond reckoning. Not only could the vehicles be operated at night almost entirely uninterrupted at a relatively high rate of speed and at a lower cost per ton mile than it

they were constantly delayed by congested traffic, but the goods could be delivered much more rapidly and the freight which would be delivered in the day from well-located distribution points would suffer very much less delay than is now the case in endeavoring to transport freight long distances through almost impassable traffic.

This subject is one that undoubtedly deserves much more careful consideration from steamship officials than it has as yet apparently received. There can be no question but what a large amount of money can be saved not only by improved methods of collecting and distributing freight from marine terminals, but also by the proper use of mechanical appliances of various kinds for handling miscellaneous freight at the steamship piers. The latest types of coastwise and seagoing ships are well arranged for handling freight, so that there is little chance for much, if any, saving in the designing and equipping of these vessels. The great field, therefore, in which steamship owners can save more money is at their terminals by reducing the cost of handling the freight. During the past two or three years we have published a score or more of articles describing many installations of freight-handling appliances that have proved to be extremely useful to steamship owners. From the diversity of these various installations, there can be no doubt that almost any type of terminal arrangement can be greatly benefited by the proper use of these appliances.

Repair Costs of Motor Ships

OWING to the reluctance of owners and builders of motor ships to give out figures showing the costs of repairs to the machinery of this type of vessel, and the length of delays required for making such repairs, the belief has been quite general that the costs and delays due to repairs must be excessive as compared with similar costs and delays with steamships. This reluctance, however, has probably been due to the fact that, as motor ships are a comparatively new departure, owners and builders have felt that figures covering only a few months' operation would furnish no reliable criterion on which to base such data. But as time has gone on, and a considerable number of successful motor ships have been in continuous service for several years, data have accumulated which can be relied upon with some degree of certainty to represent true conditions in the operation of motor ships.

A case in point is that of the Royal Packet Company, which has four motor ships operating, together with a number of steamers, all of which carry passengers, mail and freight, principally the latter. Figures covering the costs of repairs and delays due to repairs on these vessels have been given in a report from the company's engineer-in-chief at Weltevreden, Batavia, Java, where the vessels operate. In this report it is stated that the motor ship *Sembilan,* of 300 tons deadweight capacity and 200 horsepower, has made ninety voyages in three years, with a total cost of repairs of about $2,400 (£492), very little time being required to make the repairs. The three other motor ships owned by this company are each of 1,750 tons deadweight capacity and 1,400 horsepower. One has made sixty voyages in twenty months, with a total cost for repairs of about $1,600 (£328). The other two sister-ships had only just entered the company's service when the report was given out, but had at that time required no repairs.

On the other hand, the total cost of repairs to one of the company's steamships of 1,400 horsepower was $1,000 (£205) for ninety voyages in twenty-two months, which is very nearly as much as that for the motor ship of the same power.

The fuel costs for the steamship, based on a coal consumption of 1.4 pounds per indicated horsepower, which is a fair average for a ship of this class and speed, amount to $117.67 (£24.1) for a daily run of 240 miles at sea. In the case of the sister Diesel motor ships, which are capable of carrying much more cargo, because of the less space required by the engines and bunkers, and also on account of the saving in weight of the machinery, a comparison can be made with the fuel consumption of the motor ship *Siberg* on her maiden voyage from England to the Dutch East Indies via Port Said. To the latter port the vessel averaged 9.9 miles per hour on 4.6 tons of oil fuel and 2.25 tons of coal for the auxiliaries. Therefore her total daily fuel bill—oil and coal—based on daily runs of 240 miles amounts to $45.62 (£9.35). In the sister ships of this vessel the donkey boilers are fired by the exhaust gases from the main engine, which makes a further reduction in the fuel costs.

Apart from the absence of stand-by charges in port, it is estimated that with the motor ships a saving of $21,594 (£4,430) on the fuel bill alone is realized in 300 days at sea. The total saving afforded by the motor ship, however, can only be appreciated when it is realized that several hundred tons more cargo can be carried and, with the present high freight rates, this is likely to mean an even greater gain than the saving on the fuel bill.

In the case of another motor-driven cargo vessel of 600 tons deadweight capacity and 250 indicated horsepower at 200 revolutions per minute, placed in service in 1910, the vessel had run a total of approximately 80,000 miles up to June, 1915. Repairs to the main engine during this time have averaged about $500 (£102.5) per year and $100 (£20.5) per year for the auxiliaries. The total time lost by the ship has been from three to four weeks per year, which included dry docking for scraping her bottom, painting, etc.

From the foregoing, therefore, it is evident that, while against the large savings in fuel costs and in additional freight carried in a motor ship there must be charged a slightly increased repair expense as compared with a steamship, also slightly higher wages to the engineer and the addition of several extra oilers in place of the stokers required on a steamship, nevertheless the question of repair costs should be no deterent to the successful operation of motor ships. Due to the present market conditions, motor-driven ships will show to great advantage for the Atlantic trade. The best steam coal is now quoted at Cardiff and Newcastle at from $4.50 (18/9) to $6 (1/0/10) per ton, and is not likely to become lower for many years. In fact, it is extremely unlikely that it will ever get back to its old figures, because the wages of the miners cannot be expected to drop back to the comparatively low figures of a year ago. On the other hand, a motor ship can obtain oil fuel in America and carry sufficient in her bunkers for the voyage to Europe and back and still carry several hundred tons more cargo than a coal or oil-fired steamer of the same displacement could carry with bunker capacity for one voyage. These facts are fully realized by European shipowners, and to-day there are 35 Diesel engine-driven merchant ships on order, 96 percent of which will be equipped with either Burmeister & Wain or Werkspoor engines. The Western Australian Steamship Company recently paid $101 (£20.7) per ton for the new motor ship *Lalandia* of 6,700 tons deadweight capacity, a record price for a tramp ship, yet a net profit of over $125,000 (£25,600) a year is expected from this ship even after all expenses and depreciation have been paid.

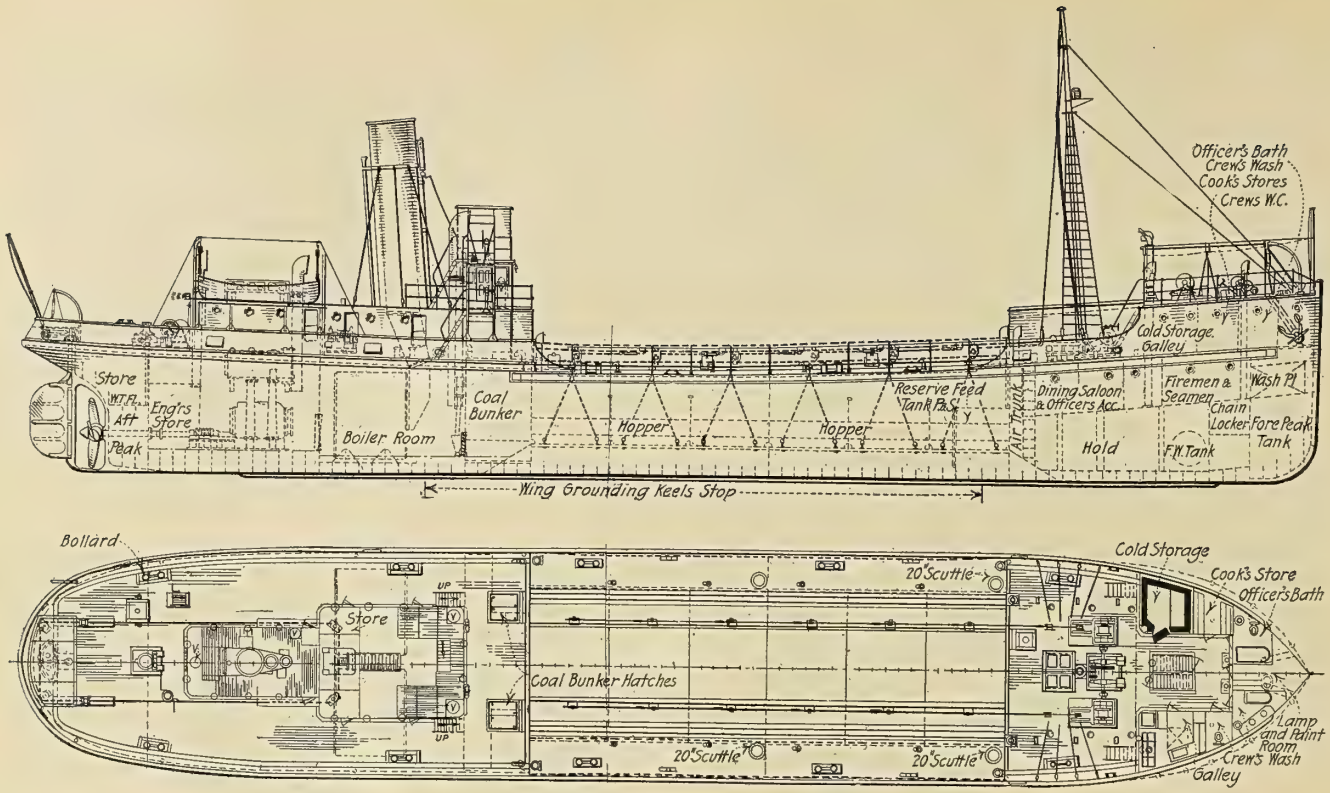


Fig. 1.—Profile and Deck Plan of Canadian Government Hopper Barge No. 2

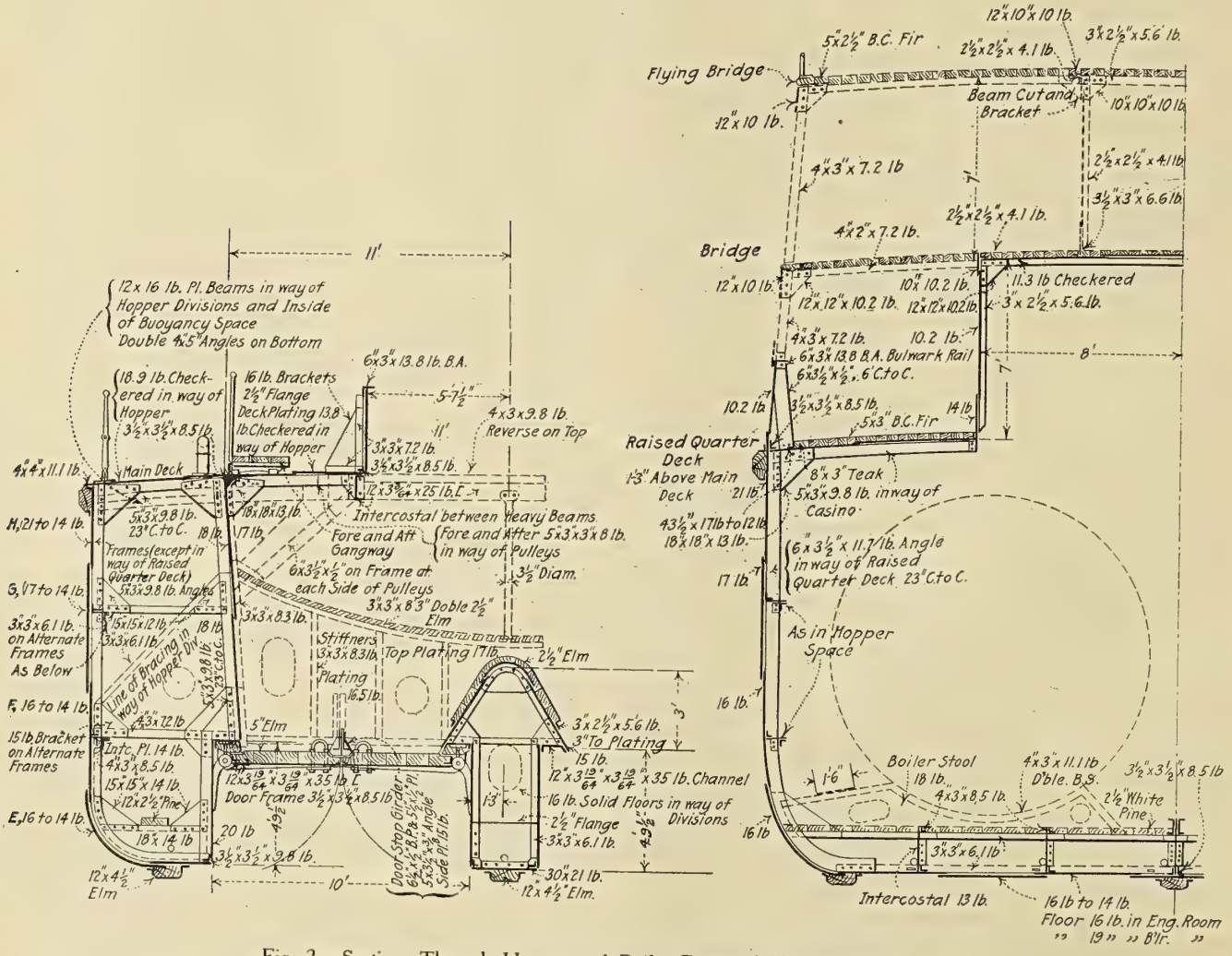


Fig. 2.—Sections Through Hopper and Boiler Room of Hopper Barge No. 2

Canadian Government Hopper Barge No 2

Self-Propelled Hopper Barge Built by the Collingwood Shipbuilding Company for Government Work on the St. Lawrence River

A hopper barge of 530 cubic yards capacity, designed for work on the lower St. Lawrence River, where she will take away spoil from both the bucket and suction dredges, was recently completed by the Collingwood Shipbuilding Company, Collingwood, Ont., for the Dominion Government. The vessel is built of steel throughout to Lloyd's highest class, and is of the following dimensions:

Length overall	187 feet.
Length between perpendiculars	180 feet.
Breadth, molded	32 feet.
Depth, molded to main deck	14 feet 6 inches.
Depth aft to bottom of grounding keels	13 feet.

The hopper space, which contains 530 cubic yards, is divided up as shown on the accompanying plans. There are six sections of the hopper in all, and each section is

Fig. 3.—Hopper Barge No. 2

provided with two doors constructed as shown on the midship section.

These are secured by very heavy cast steel strap hinges and are lowered and raised by means of 1¼-inch bridal chains, having heart-shaped shackles at the upper end to take up any movement. A chain leads up through the deck casting and over a sheave and is attached to a strong draw-bar running the full length of the hopper coaming. Two threefold purchase blocks are fitted at the fore end and a wire rope is led from these on to the drum of a powerful winch situated at the after end of the forecastle.

The port or starboard doors can be raised or lowered independently, and any set of doors can be raised or lowered without interfering with the other. When the doors are hove up they are set up by heavy cotters at the deck casting. The doors are made of rock elm and the hopper keelson is lined with similar material.

As will be seen from the midship section, the doors are recessed well into the hull to enable the vessel to be put ashore at any convenient place and the doors lowered for repairs without projecting beyond the bottom of the ship.

Accommodation for a crew of fourteen is provided in a 'tween-deck forward, the rooms being spacious, well lighted and ventilated.

The machinery consists of a set of compound surface-condensing engines made by the builders, having cylinders 20 inches and 40 inches by 27 inches stroke, and designed to develop 700 indicated horsepower at 120 revolutions, and 125 pounds steam-pressure. The boilers, two in number, constructed by the builders, are of the regular Scotch type, 12 feet 3 inches diameter by 10 feet 6 inches long, working under natural draft.

A very complete set of auxiliaries is installed, including Weir's feed pump, Gwynnes' circulating pump, feed heaters, filters, etc., and everything has been arranged so the vessel can work in either salt or fresh water. A very complete installation of electric light has been provided, and sanitary and heating arrangements are provided suitable for a vessel of this class.

Trials were run on the measured mile at Owen Sound on September 8, when a mean speed of 8¼ knots was obtained, and, on a six-hours' trial, 750 horsepower was developed with the vessel loaded with spoil to a mean draft of 13 feet. Trials of the dumping gear were carried out later, and the gear worked smoothly. Mr. Charles F. M. Duguid, naval constructor of the Department of Marine and Fisheries, witnessed the trials on behalf of the Dominion Government, the builders being represented by Mr. John S. Leitch, manager of the company.

The vessel sailed for Quebec on September 11.

Shallow Draft Boats and Barges for River Navigation*

BY COL. L. H. BEACH AND LIEUT.-COL. CHAS. KELLAR

The advance which has been made in the size, machinery and power of ocean-going vessels within the past fifty or sixty years has been almost marvelous. Vessels have increased from a gross tonnage of 2,860 tons in 1850 to 43,500 tons in 1910. Allowable boiler-pressure has increased from 25 pounds in 1857 to 200 or 300 pounds at this day. The length of vessels has steadily increased from 282 feet to 500 and 600 feet for freight-carrying vessels and 850 feet for passenger and quick-freight traffic. No advance worthy of the term has, however, been made in river craft, which follow the same method of construction and practically the same methods of propulsion as were in vogue fifty years ago. The only improvement made is the allowable boiler-pressure.

To see whether developments could not be made which would be commercially economical and correspondingly advantageous to the community, the Congress of the United States recently directed the design and construction of "two experimental towboats of modern but different types with a complement of suitable barges and necessary loading and unloading facilities for towing and delivering supplies along the Mississippi River and its tributaries." A board of engineer officers was appointed to make the necessary investigations and carry out the work involved. It found nothing in print or otherwise of record in available form connected with its task, and has had to investigate the entire field practically from the beginning. It has made thousands of tests concerning the best forms for barges and the most economical methods of grouping in tows, and finds that while the form of barge giving best results is to a certain extent dependent upon the duty which is to be performed—that is, whether it will be used for a long or short haul, whether towed always singly or in combination with others in a tow—there are

* Abstract of paper read before Engineering Section of Second Pan-American Scientific Congress, Washington, D. C., January 3.

two or three forms which give best results, and that other shapes should be excluded if the greatest efficiency is desired. One result of their work is to decide the question which had been greatly debated by coal producers in the upper Ohio and its tributaries as to whether economical results could be secured by using sets of nestable barges, the small barges to be towed up stream empty within the larger barges. The board's results enabled a definite answer to be given to this problem.

The board further conducted a most extensive series of experiments concerning paddle wheels upon which nothing definite had previously been determined and concerning which no experiments had been scientifically conducted. Some of the facts developed by the board's experiments are as follows:

A feathering wheel has practically the same power as a radial wheel of twice the diameter.

The thrust increases with the immersion when width of blade and number is constant.

The thrust increases with the increase in number of floats within usual limits, and varies approximately directly as the length of float.

No additional thrust is gained by separating the wheel into two parts. Thrust varies approximately as diameter for constant immersion.

The efficiency decreases rapidly toward the higher values of slip and also with increase of immersion.

The thrust and efficiency of a feathering wheel vary with the position of the feathering eccentric, the thrust generally decreasing with the increase of eccentricity.

The Floating Hospital Helen C. Juilliard

New Vessel Built from Designs by Tams, Lemoine & Crane for St. John's Guild, New York

The new floating hospital for the St. John's Guild, New York, to replace the *Helen C. Juilliard,* long known in New York harbor, and to be called by the same name, will be a great improvement over the old ship, which has done such good work for so many years.

The new boat has been built from Messrs. Tams, Lemoine & Crane's designs and under their supervision by the American Car and Foundry Company, Jackson & Sharp plant, Wilmington, Del. Her length on deck is 230 feet 8 inches, breadth outside the planking of 45 feet, and draft aft 10 feet.

She is of wooden construction, all materials going into her being most carefully selected. In this connection it may be stated that the architect's inspector—Mr. J. J. Driscoll—was the supervising inspector of the old ship.

Her keel is of white oak and measures 12 inches by 14 inches. The stem is of white oak, fastened to the keel with hook scarphs. The apron is of white oak, as are also the stern and rudder posts, the deadwoods being of yellow pine.

The frames are of white oak and chestnut, double sawn, spaced 24 inches apart. The upper part of the frames extends clear up to the hurricane deck. The floors are of oak, sided 5 inches, and the keelsons are of yellow pine, of which there are seven, running the full length of the vessel for strengthening fore and aft.

In addition, there are two bulkheads running fore and aft, of yellow pine, on top of which are set the hog frames. Stringers and clamps are also of the same, and the knees are hackmatack. The deck planking is of yellow pine.

The principal details of the hull construction and the main scantlings are shown in the midship section, Fig. 1.

The deck plans show the arrangement of the ship, which has been very carefully thought out. The dining room has tables seating four hundred people. There is a waiting room, examination room and operating room, which in every detail will be up to date. There is also a large ice box, a sterilizing room, four distinct wards, and on the hurricane deck is an isolation house.

Fresh-water tanks of 15,000 gallons capacity are fitted, in addition to salt-water tanks of 10,000 gallons capacity, permitting patients having clear salt baths at all times. In all shower, bath and toilet rooms monolithic floors are laid, all sanitary details of this sort being of first importance.

The fire-fighting system has been most carefully worked out, and consists of fire pumps of large capacity with outlets serving all parts of the ship. For supplies there are

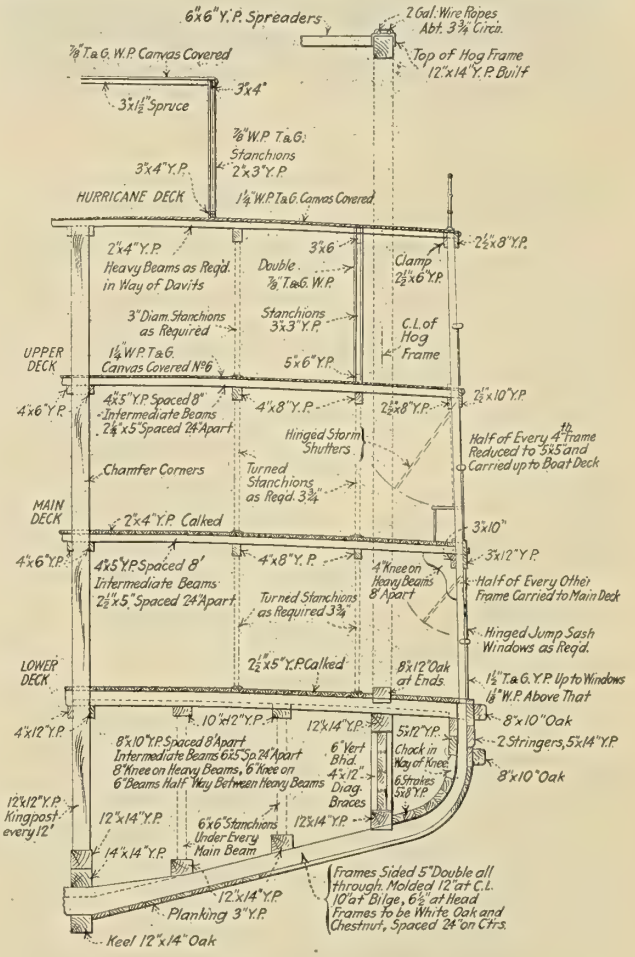

6"x6" Y.P. Spreaders
2 Gal. Wire Ropes Abt. 3¾ Circ.
⅞" T.&G. W.P. Canvas Covered
Top of Hog Frame 12"x14" Y.P. Built
3"x4"
3"x1½" Spruce
⅞" W.P. T.&G. Stanchions 2"x3" Y.P.
3"x4" Y.P.
HURRICANE DECK
1¼" W.P. T.&G. Canvas Covered
2½"x8" Y.P.
2"x4" Y.P. Heavy Beams as Reqd. in Way of Davits
3"x6"
Double ⅞" T.&G. W.P.
Clamp 2½"x6" Y.P.
3" Diam. Stanchions as Required
Stanchions 3"x3" Y.P.
C.L. of Hog Frame
UPPER DECK
1¼" W.P. T.&G. Canvas Covered No. 6
5"x6" Y.P.
4"x6" Y.P.
4"x5" Y.P. Spaced 8" Intermediate Beams 2½"x5" Spaced 24" Apart
4"x8" Y.P.
2½"x8" Y.P.
2½"x10" Y.P.
Hinged Storm Shutters
Half of Every 4th Frame Reduced to 3"x5" and Carried up to Boat Deck
Chamfer Corners
Turned Stanchions as Reqd. 3¾"
MAIN DECK
2"x4" Y.P. Calked
3"x10"
3"x12" Y.P.
4"x6" Y.P.
4"x5" Y.P. Spaced 8' Intermediate Beams 2½"x5" Spaced 24" Apart
4"x8" Y.P.
4" Knee on Heavy Beams 8' Apart
Half of Every Other Frame Carried to Main Deck
Turned Stanchions as Required 3¾"
Hinged Jump Sash Windows as Reqd.
2½"x5" Y.P. Calked
8"x12" Oak at Ends
LOWER DECK
1½" T.&G. Y.P. Up to Windows ⅛" W.P. Above That
8"x10" Oak
4"x12" Y.P.
10"x12" Y.P.
12"x14" Y.P.
5"x12" Y.P.
2 Stringers, 5"x14" Y.P.
8"x10" Y.P. Spaced 8' Apart Intermediate Beams 6"x5" Spaced 24" Apart 8" Knee on Heavy Beams, 6" Knee on 6" Beams Half Way Between Heavy Beams
6" Vert. Bhd. Chock in Way of Knee
4"x12" Diag. Braces
6 Strakes 5"x8" Y.P.
8"x10" Oak
12"x12" Y.P. Kingpost every 12'
12"x14" Y.P.
6"x6" Stanchions Under Every Main Beam
12"x14" Y.P.
14"x14" Y.P.
12"x14" Y.P.
Frames Sided 5" Double all through. Molded 12" at C.L. 10" at Bilge, 6½" at Head Frames to be White Oak and Chestnut, Spaced 24" on Ctrs.
Planking 3" Y.P.
Keel 12"x14" Oak

Fig. 1.—Midship Section

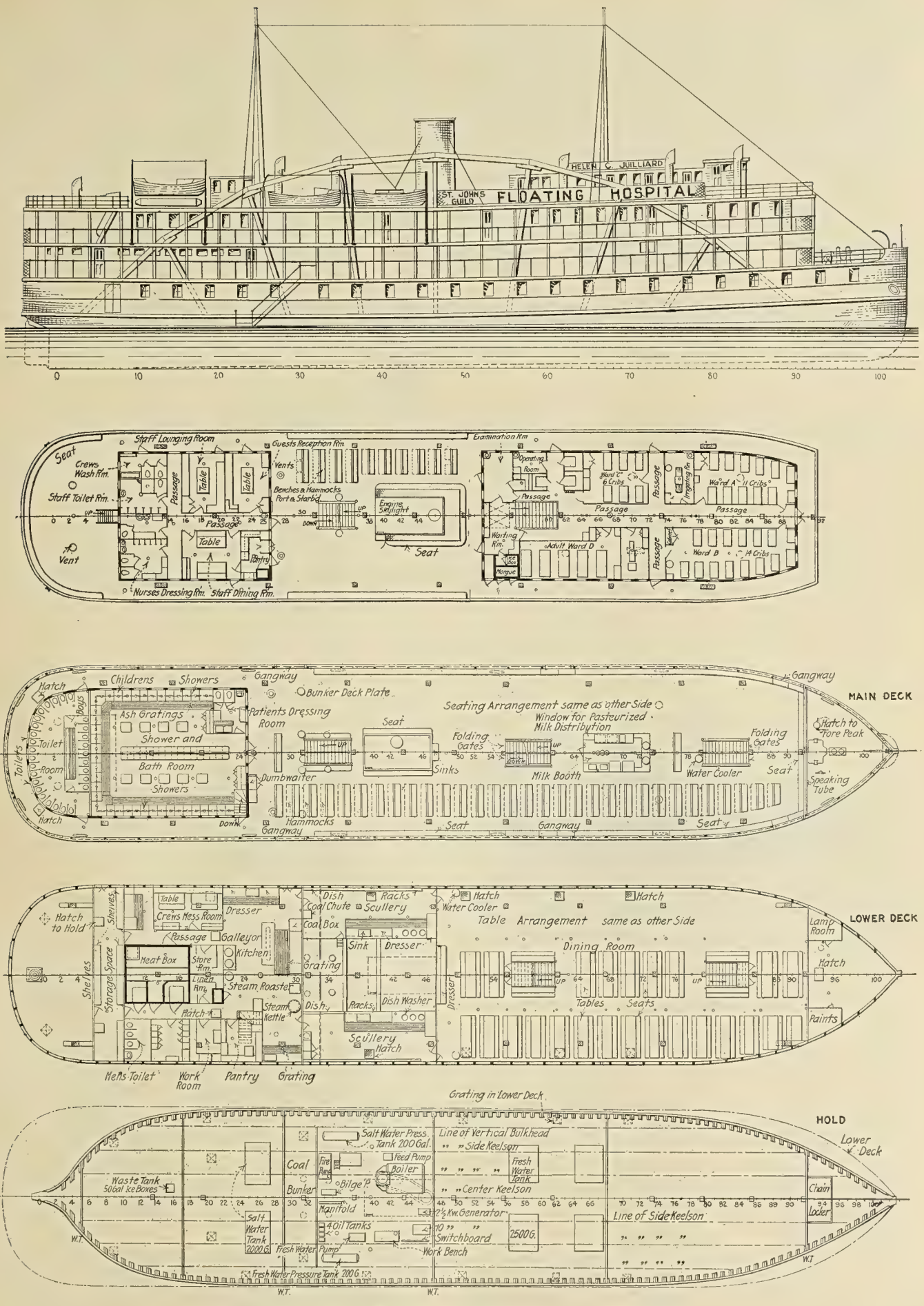


Fig. 2.—General Arrangement of Floating Hospital *Helen C. Juilliard*

large refrigerating rooms, with separate meat rooms, vegetable and dairy rooms, the latter being particularly large to accommodate the large supply of milk necessary.

A Scotch boiler 7 feet in diameter and 9 feet long is installed for running all the auxiliaries, the electrical equipment being most complete, including two of the latest type generators for supplying, in addition to the lighting system, current for operating fans, motors and thermostatic pads for use in the wards. The boat itself will not be self-propelled, but towed, as in the case of the former ship.

The life-saving equipment includes four 18-foot and two 20-foot life boats, also an 18-foot yawl, life rafts, life preservers, etc. All this equipment is based on the capacity of sixteen hundred.

There are living accommodations for doctors, nurses, attendants and crew, with separate dining and sitting rooms.

New Tugs for Central Railroad of N. J.

Extensive Experience in Harbor Lighterage Service and Towing Leads to New Features and Improvements in Design of New Tugs

BY ALTON A. NORTON

Two new tugs have recently been built for the Central Railroad of New Jersey which embody new features and improvements which have been brought about through experience with former tugs in the service of the same company.

One, the *Bethlehem,* was built by the Staten Island Shipbuilding Company, of Port Richmond, N. Y., for the lighterage service. It was deemed advisable to use a high-pressure engine on account of the muddy rivers and creeks they are compelled to navigate. The engine is 22 inches by 28 inches, and steam is supplied at 150 pounds gage-

Fig. 1.—The *Bethlehem*

pressure by a three-furnace Scotch boiler 14 feet 6 inches diameter. Principal dimensions of the *Bethlehem* are as follows:

Length on deck	98 feet 0 inches
Beam, molded	24 feet 0 inches
Depth, molded	12 feet 7 inches
Draft, loaded	11 feet 0 inches
Displacement	about 320 tons

Except in size, the general arrangement of the two tugs is practically the same. The other tug, the *Newark,* was built at Elizabethport, N. J., by John W. Sullivan, and the machinery installed at the Ninth street yard of the same company.

The following is a general description of the *Newark,* which is being used in the car-float service of the company:

General Dimensions

Length overall	110 feet 0 inches
Length between perpendiculars	97 feet 9 inches
Beam, molded	26 feet 0 inches
Depth, molded	14 feet 6½ inches

Hull Particulars

The hull is of steel, built exceptionally strong. In the design care has been taken to make the interior accessible for efficient scraping and painting. As will be seen in the accompanying profile, her forefoot and deadwood

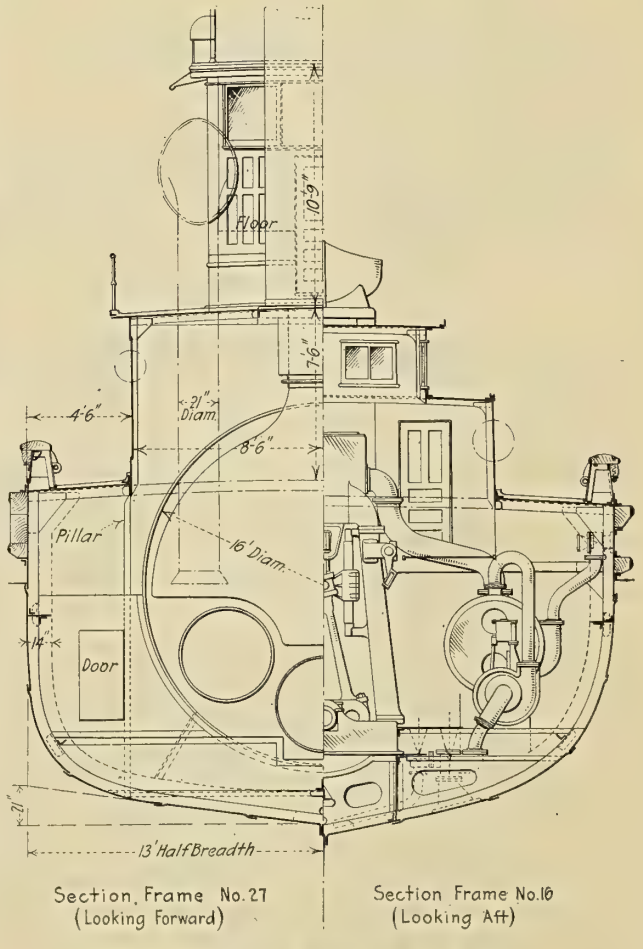


Fig. 2.—Midship Section of the *Newark*

aft have been cut away to a coniderable extent. This enables the tug to turn in a circle of shorter radius. A balanced rudder of the side-plate type is fitted.

Reverse frames follow the top of floor plates only and are continuous athwartships. They are doubled in way of the engine room.

A deep keelson extending from the collision bulkhead to the after-end of the engine room is fitted. Longitudinal strength is further assisted by side keelsons and stringers, the side stringer being deep and formed of intercostal plates and clips, between frames, with a continuous angle along the outside of the frames. This gives great stiffening power to plating below the lower fender rail, where it is very frequently needed.

The bulwark construction is clearly shown in the section. The bulwark aft has been set in, and three heavy cast steel protecting chocks are fitted. This arrangement was adopted to minimize the chances of breaking the bulwarks at this point.

The space between the first and second fender guards is filled in with solid wood from the stem to amidships, the object being to reduce the damage to the hanging fenders caused by the localized bearing of the fender guards on them.

A cross coal bunker is fitted between the engine and boiler room, with a coaling trunk extending down from the upper deck. This is a very convenient method, as the tug can take on her full fuel capacity without having to turn around at short intervals on account of excessive list, as encountered in filling side bunkers. Two ash bunkers are also fitted in the after corners of the fire room. A passageway between the boiler and engine rooms is cut through the cross coal bunker. Boiler feed water is supplied from two built-in fresh-water tanks, the forward one having a watertight flat forming the top with the frames bracketed to this.

There are four watertight bulkheads placed as shown in the profile. The deck is of steel, covered with "Litosilo."

Deck House

The deck house has been very conveniently arranged, as will be seen from the deck plan. The archway cut through the house is a novelty and has proven to be very handy in the working of the tug. It affords quick access from one side to the other, also a temporary shelter for the deck crew. Lines are conveniently stowed in the archway, where they dry quickly and are immediately accessible. Entrances to the engine room are fitted through the after part of the archway, the working platform of which is lowered below the main deck.

The primary purpose of lowering and narrowing the after part of the deck house was to obtain an unobstructed view of the stern from the pilot house. To further this purpose the smokestack and ventilators were made elliptical in section. It has been found that this form of engine room gives better light, better ventilation and also a better view from the handling platform. The deck house is constructed of steel and the pilot house of wood.

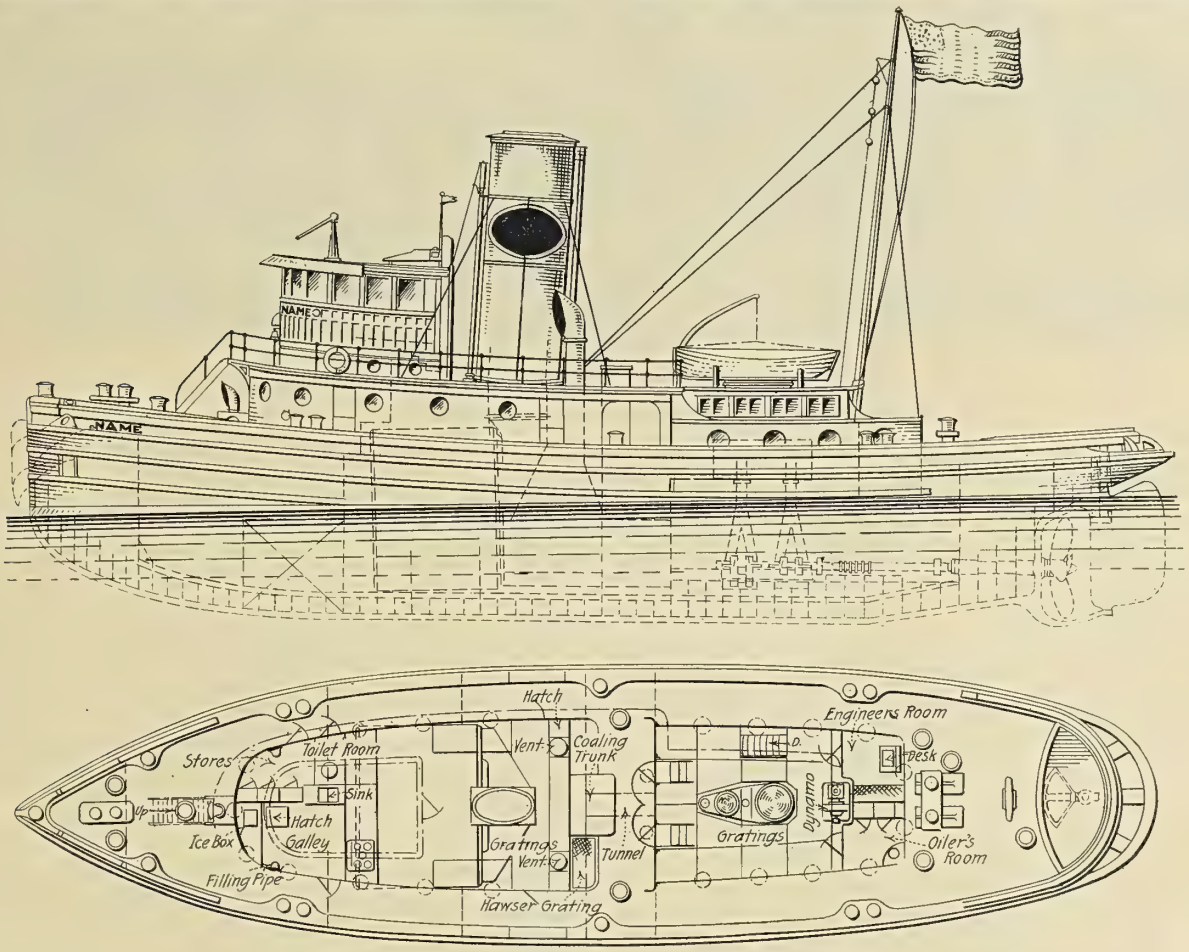


Fig. 3.—General Arrangement of the *Newark*

Machinery

The propelling machinery consists of one fore and aft compound engine, one cylindrical return tube boiler, a surface condenser with centrifugal circulating pump, independent air, feed, bilge, fire and sanitary pumps, feedwater heater, electric lighting plant with direct-connected engine.

Fig. 1.—Bunkering a Steamship at New Southern Railway Coaling Pier

The cylinders of the main engine are 19 inches and 40 inches diameter by 30 inches stroke. High-pressure valve is a semi-plug piston type, and the low-pressure cylinder is fitted with a double-ported slide valve. The valves are operated by Stephenson link-valve gear.

The feed-water heater is of the multicoil type, the exhaust from all auxiliary machinery being run into it to supply the necessary heat.

The air pump is a vertical twin-beam, single-acting pump; the feed and bilge pumps are of the vertical, simplex type, and the fire pump of the horizontal duplex type.

The boiler is 16 feet inside diameter by 12 feet 3 inches long over the bottom heads, with a working-pressure of 175 pounds per square inch. The three suspension furnaces are connected to a common combustion chamber. The boiler is also fitted with a modern form of circulator.

The steering gear is of the ram type as manufactured by the Staten Island Shipbuilding Company.

The tugs are an excellent example of the progress made in the marine departments of the various railroads, due to their use of one particular type of boat. By close observation of previous tugs in service and careful recording of data, new tugs are designed that embody the good features and eliminate some of the difficulties of former tugs.

The specifications require many parts to be constructed in accordance with Central Railroad of New Jersey standards, designs which are the result of experience in former boats.

The hull and general arrangement were gotten out by Mr. W. S. Macdonald, marine inspector of the Central Railroad of New Jersey, and the machinery equipment by Mr. William Riechman, the chief engineer of the marine department.

LIGHTSHIP LAUNCHED.—Lightship *No.* 101, under construction at the works of the Pusey & Jones Company, Wilmington, Del., was launched on January 12.

SALE OF DANISH VESSELS PROHIBITED.—The Danish Government has forbidden the transfer of Danish vessels to foreign ownership.

Southern Railway Coaling Pier at Charleston, S. C.

The first modern coaling pier on the Atlantic seaboard south of Norfolk has recently been completed by the Southern Railway at Charleston, S. C., at a cost of $500,000 (£102,500), although the plans provide for the ultimate development of the plant to four times its present capacity. The present plan consists of a steel pier 375 feet long, a 3,065-foot approach trestle, a 370-foot tail track and a 2,250-foot trestle for the return-empty track. On the pier is a movable coal-loading tower with a hopper of 200 tons capacity into which cars can be dumped at the rate of one every two minutes, and from which the coal is transferred to the vessels at the pier by a link

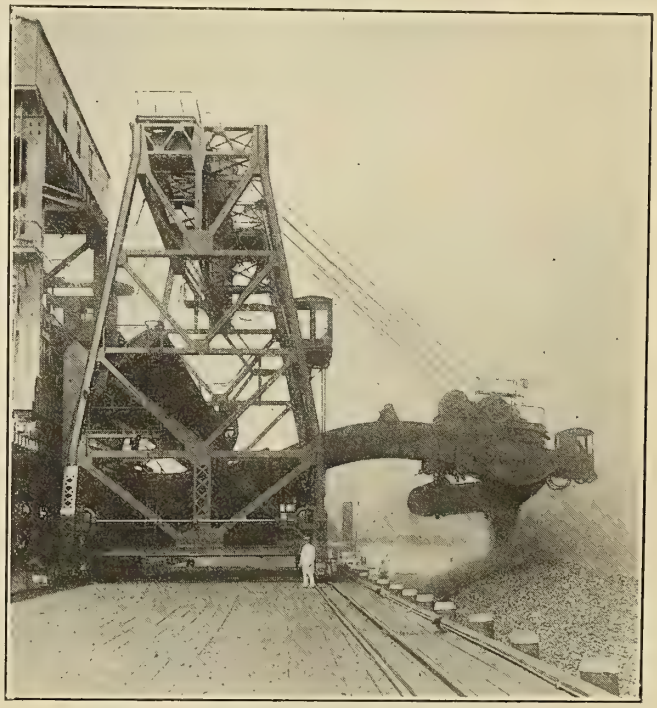


Fig. 2.—Coal-Loading Tower in Operation

belt conveyor operating in a movable boom which can be adjusted in accordance with the position of the coaling hatches on the vessel. The cars are pushed onto the car dumper by switch engines, and after they are dumped they pass down a 2½ percent grade to a kick back that gives them the necessary momentum to carry them down a 1 percent grade on the empty return track.

The car dumper and loading tower, including all machinery and structural steel, were furnished by the Wellman, Seaver, Morgan Company, Cleveland, Ohio. The steel framework of the car dumper extends 56 feet 9 inches above the top of the approach trestle and the dumping cradle is operated by two 150-horsepower electric motors arranged in parallel located at the top of the structure. The cradle is moved along the trestle track by a 100-horsepower motor, all of the operations being controlled from a central station.

The loading tower, which is at a level 17 feet 2 inches below that of the car dumper, carries a hopper with fixed sides and back, but with a movable bottom or pan pivoted at the lower end. Normally this pan is on a slope of about 45 degrees, permitting it to be emptied by gravity, but when it is desired to shift the tower independent of the dumper, the pan can be put in a horizontal position, doubling its capacity from 100 to 200 tons. The coal is taken from the hopper by a belt conveyor, operating in a curved boom having a maximum reach of 30 feet 11 inches beyond the pier. The end of the boom is equipped with a telescopic chute which may be swung in an athwartships direction, and at the end of which is a trimmer cowl which swivels on a vertical axis, thus giving extreme flexibility for placing the coal in the vessel and reducing hand trimming to a minimum. All movements of the loading tower and boom are actuated by electric motors controlled from two stations.

Bulk Cargo Steamer Chokyu Maru

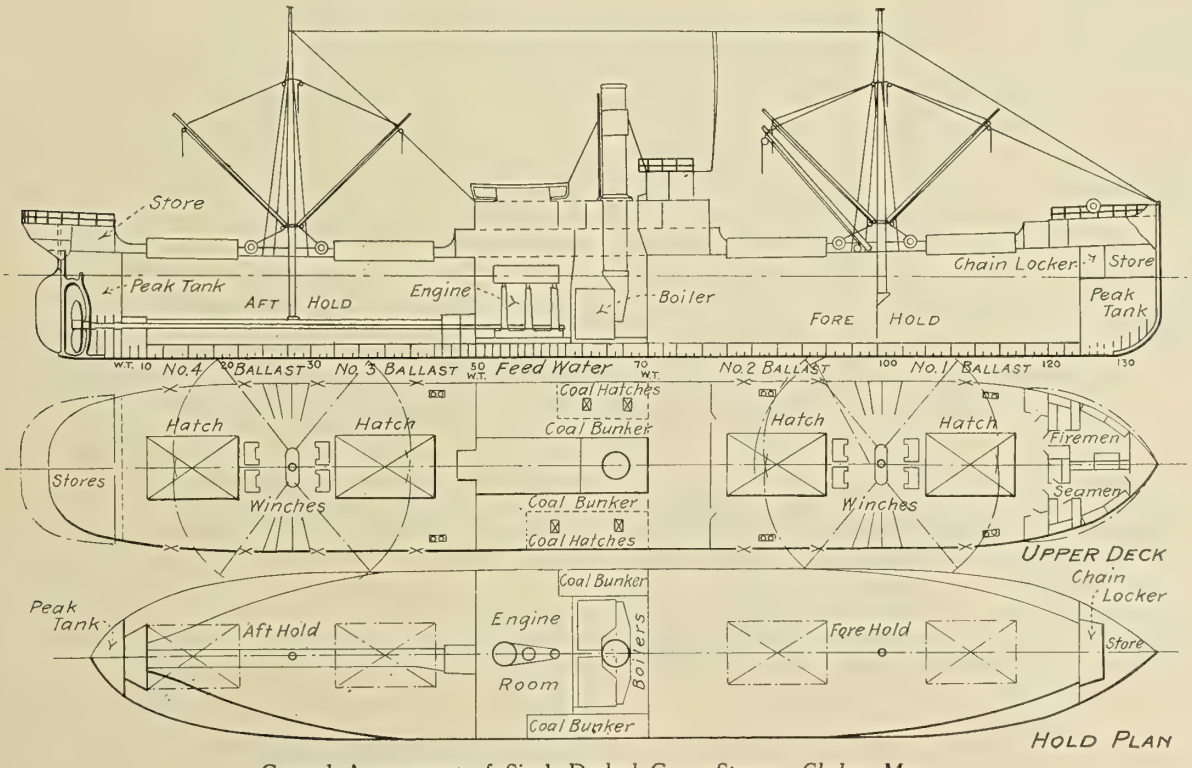
Description of New Japanese Cargo Steamer Built by the Uraga Dock Company—Launching and Trial Data

BY T. RYU

The single-decked cargo steamer *Chokyu Maru* was recently completed by the Uraga Dock Company, Japan, for the Kamaishi Steel Works, to transport iron ore and coal between Formosa and Japan. The principal dimensions of the vessel are as follows:

Length overall	277 feet 7 inches
Length between perpendiculars	268 feet
Breadth molded	40 feet 9 inches
Depth molded	23 feet 6 inches
Draft, loaded	19 feet 9¼ inches
Displacement	4,887 tons
Carrying capacity	3,067 tons
Speed	11 knots
Indicated horsepower	1,060

As shown in the general arrangement plans, the lengths of the deck erections are shortened so as to afford longer



General Arrangement of Single-Decked Cargo Steamer *Chokyu Maru*

hatchways, making the forecastle 27 feet, the bridge 63 feet and the poop 23 feet long. No. 1 and No. 4 hatches are 22 feet by 15 feet and No. 2 and No. 3 hatches 24 feet by 15 feet, so that the length and width of the hatch openings is 34 percent of ship's length and 37 percent of ship's breadth respectively.

Two 4-ton winches, with 7-inch cylinders and 12-inch stroke, are provided for each hatch, while a 15-ton derrick at No. 2 hatch serves to handle heavier loads. The height of the hatch coamings is 3 feet, and they are stiffened with bulb angles.

Hollow cylindrical pillars are adopted in place of extra strong hold pillars, in order to lessen the wear of wire ropes in cargo handling. No ordinary ladders are provided for in the hatchways, but they are fitted inside the masts so as to give clear space in the holds. Hold stringers are dispensed with by using thicker outside plating.

Taking coal as cargo at 45 cubic feet per ton of 2,240 pounds, the capacity of the fore hold is 1,806 tons and of the aft hold 1,261 tons. The side coal bunkers contain 124 tons and the reserve bunker in the bridge contains 218 tons, making a total of 342 tons.

A skeleton floor system is adopted for the inner bottom plating, extending from bilge to bilge. The capacities of the peak tanks and the double bottoms are arranged so as to avoid excessive trim at light draft, and to have about one foot trim without ballast at load draft. The capacities of Nos. 1, 2, 3 and 4 water ballast tanks, and the fore and after peak tanks are 79.2, 94.7 and 112.8 tons of water respectively. Under the engine room 45 tons feed water is carried.

The construction of the hull adopted has reduced the weight of steel required by about 3 percent as compared with ordinary methods of construction. The weight of the vessel is 1,370.3 tons, consisting of 1,155.8 tons for the hull and 214.5 tons for the machinery, including the water in the boiler.

Launching Data

Declivity of keel blocks	1/17
Declivity of launching ways	1/16
Length of sliding ways	217 feet
Width of sliding ways	1 foot 9 inches
Width of standing ways	2 feet
Center to center of sliding ways	14 feet 6 inches
Average pressure on standing ways	1.35 tons
Maximum pressure on the fore poppet when the stern lifts	315 tons
Height of water at the end of standing way,	5 feet 6 inches
Launching speed	16.2 feet per minute
Launching weight, including cradles	1,033 tons
Average draft when afloat	4 feet 7⅛ inches
Displacement when afloat	971 tons
Center of gravity of the hull and cradles,	0.83 foot aft of amidships

Machinery

Propulsion is by a triple expansion engine with cylinders 19 inches, 31½ inches and 52 inches diameter by 39 inches stroke, operating at 75 revolutions per minute. The cooling surface of the condenser is 1,207.5 square feet. Steam is generated by two Scotch boilers 12 feet 6 inches diameter by 10 feet 4 inches long at a working pressure of 180 pounds. The propeller is 14 feet diameter, 4-bladed, with 17 feet 3¾ inches pitch.

Trial Data

Weather	Windy and rain
Average draft	9 feet 7¾ inches
Trim	5 feet 7 inches
Displacement	2,199 tons
Midship area	374.8 square feet
Average speed	11.02 knots
Indicated horsepower	1,161.6
Slip of propeller	17.3 percent
$\frac{D^{2/3}\,V^3}{\text{I. H. P.}}$	196.79
$\frac{V^3}{\text{I. H. P.}}$	436.16
$\frac{\text{I. H. P.}}{\text{I. H. P.}}$	0.318
$\frac{\text{H. S.}}{\text{I. H. P.}}$	10.95
G. A.	
Coal consumption (Australian coal),	3.1 pounds per horsepower hour

Harbor and Port Terminal Facilities and Work*

BY H. MC L. HARDING†

To create a satisfactory interchange of commerce between the States of the Pan-American Union it is necessary not only that there be a large merchant marine for the carriage of goods and commodities between ports, but that there should be such harbor terminal facilities and works as will secure the greatest speed and economy in cargo transference (loading and unloading) between vessels and shore, and in the handling (assorting, distributing and tiering) upon the shore, not only equal to but superior to any harbor works and equipment in the world.

The paper treats of the following subjects: Influence of terminal costs, commercial supremacy, terminal facilities, ships' detention, rapidity of transference due to terminal design, similarities in port designs, relation of terminal expense to carriage expense, typical terminal design, rail and water coördination, congestion points, quay terminals, transit shed, warehouse, no railway cars in dray shed, types of mechanical appliances, railway connections, industrial and commerce terminals, comprehensive plan in the beginning, terminal elements, freight movements, terminal units, size of units, the substructure (quay walls), fresh-water quay terminals, relieving platform, teredo-infested waters, the superstructures, shed capacity, railroads, mechanical appliances, terminal designing.

Conclusions

First.—For all the ports of the Americas, and especially for those concerned with Pan-American commerce, those harbor works, terminal facilities, operating methods, and mechanical appliances should be adopted which have proved most efficient in respect to time and economy of cargo transference and handling.

Second.—That all new terminals for inland and ocean navigation should be so designed, planned and proportioned in respect to the piers, quays, sheds, warehouses and railway tracks as to secure easy, quick and economical inter-coördination between all these terminal elements and with the water carriers.

Third.—That not only should the world's best be adopted and then adapted to the operating conditions of

* Abstract of paper read before Engineering Section of Second Pan-American Scientific Congress, Washington, D. C., January 4.

† Consulting Engineer, New York.

each country, but terminal engineers should continually study to achieve something better.

Fourth.—That the securing of the greatest speed possible in all marine freight terminal movements is of all terminal features that which is the most essential for reducing the cost of transporting freight.

Cooperation Between Motor Trucks and Steamship Lines in Handling Freight*

Territory for Collecting Freight Increased Four Times by Motor Truck Service—Greater Shipments of Fruit to Chicago by Boat

BY H. L. WHITTEMORE†

Over 300,000,000 tons of freight are annually hauled to and from marine terminals alone, a total of more than 600,000,000 tons handled, or 1,800,000 tons every working day. With but little exception, the entire movement of railroad and steamship freight has to be handled by some highway carrier. This fact alone establishes a working relationship between motor trucks and the common carriers.

The piers and railway freight terminals of our big centers in many cases present a problem that not only militates very strongly against the motor truck, but creates a condition that makes it impractical for the motor truck to step in and even point a possible way to solve that problem. The congestion at these terminals is so bad that motor-truck owners are loath to send their trucks there. A motor truck under most conditions is far more economical than horse-driven equipment, but a motor truck, like any other vehicle, is only making money when it is under way with a paying load, and where conditions hold it idle for a long time the interest charges on its first cost cut into its earning capacity at a rapid rate.

Congestion at New York Piers

In New York City the congestion at the piers and freight stations is terrific, and I cannot help feeling, after studying the conditions there, that the steamship and railroad lines have only begun to take the possible steps for relieving these conditions. An instance where conditions have been very materially improved is on the piers of the Fall River Line, where freight is handled on the pier by mechanical freight handlers. Before this system was installed it was necessary for all trucks either bringing freight to the line or taking it away to go on the pier. Now no trucks delivering to the line go on the pier, but simply back into the bulkhead and deposit their loads, which are handled from there by the mechanical handlers. On an average day 300 trucks deliver freight to the Fall River Line and 200 collect freight from them. Before the old system was abolished each of the 500 trucks had to work on the pier; now the same space is reserved for 200. As a result of this, shippers dealing largely with the Fall River Line find that motor trucks are paying better than where they are working at some other terminals.

It is not at all infrequent for trucks to have to wait from three to four hours every working day in the year. With motor trucks this delay causes an actual loss of from $200 to $480 (41/13/4 to 100/0/0) per vehicle per year. As methods to overcome this delay are adopted, more and more motor trucks will be used.

Looking at this question from the other side, let us see whether the increased use of motor trucks cannot affect a substantial saving in dollars and cents for the companies operating the freight terminals. The more congested the space the more the rental of the terminal costs—on the west side of New York the rental of pier space is something over 80 cents (3/4) per square foot per year. A single horse delivery wagon is about 18 feet long and occupies about 90 square feet, while a small motor vehicle of similar capacity occupies scarcely more than 10½ feet in length, or 60 square feet, a saving of at least 25 percent when an extra space for maneuvering is allowed The average saving per year of a motor truck in this one respect would be about $16.00 (3/6/8), as about the same difference exists between the two-horse team and the larger motor as betwen the smaller units. It is estimated that about 40,000 vehicles are constantly engaged in trucking to piers on the west side of New York—if only half of these were replaced by motor vehicles the annual saving to pier owners of this one district in increased space available for storage would be close to $300,000 (£61,500) a year.

An interesting case of the application of motor trucks to steamship terminal problems is the service rendered during the past summer by the Fruit Belt Auto Truck Express to the fruit growers of the fruit belt around Lake Michigan. I am particularly interested in this work because I happened to be in Chicago last June when our Chicago office sold the first trucks to this company. While this operation was only carried on during a part of last season, it has already demonstrated its practicability and has served as a beacon that lights the way to a use for motor trucks that it is impossible to see a limit to.

How Fruit Growers Benefit by Motor Truck Line

Of necessity a boat is handicapped by lack of mobility, and steamship lines have had to draw their freight business from industrial and agricultural communities within a short distance of navigable waters. An active-minded man who lived in South Haven, Mich., saw the advantages both to the steamship line and to the fruit growers if he could extend the territory from which it was economically practical to ship fruit to Chicago by boat. This man inaugurated the Fruit Belt Auto Truck Express, and, as he renders a real service to the steamship company and to the farmer, he collects from both of them. The farmer pays him a certain amount per barrel or bushel for transporting fruit to the boat, and the steamship company pays a percentage of the freight rate for the new business created for them.

During the summer of 1914 the Chicago and South Haven Steamship Company carried about 500,000 pieces

* Extracts from a paper on "The Place of the Motor Truck in General Transportation," read before the Rivers and Harbors Congress, Washington, D. C., 1915.

† With the Autocar Company, Ardmore, Pa.

of fruit freight, a very considerable item of business. All of this freight came from farmers living within a radius of 15 miles of South Haven, most of it from those within 8 or 10 miles of the lake. The motor trucks easily reach farms 25 to 30 miles away and really extend the possible territory about four times. Although I have not the actual figures of increase, I have a letter from the steamship company assuring us that any steps we might take to increase the service during the coming season would meet with their heartiest coöperation.

It is easy to see in this case how motor-truck service is a distinct advantage to the farmer and to the steamship line. The farmer is saved the inconvenience of hauling to a terminal, and in addition to that has his fruit carried to Chicago in a few hours in the cool of night, so that the fruit reaches the market in better condition. Moreover, the produce gets to Chicago much earlier in the morning and arrives at a point much nearer to the commission houses to which it is consigned. The shipments are sorted on the dock according to the consignees and a bulk receipt taken for all the fruit destined for each consignee, which greatly facilitates delivery.

Advantages of Motor Truck Service

From the point of view of the steamship company this system has two very decided advantages. It opens up new business that they previously were unable to get and it reduces the overhead expenses of handling business, as the consignments come to them in bulk instead of in smaller quantities from each individual shipper.

The most conclusive evidence that this sales plan has all the elements of a permanent success is the expression from the farmers themselves. The Chicago and South Haven Steamship Company have forwarded to us several letters from fruit growers that testify to their appreciation of the service rendered. A. L. Stewart, whose address is Route 1, Box 50, South Haven, writes: "The Autocar trucks running on the main roads and leading to the steamboats which run between South Haven and Chicago are so great an advantage to the farmers that every effort should be made to promote their use. Several fruit growers have signed their names to the following letter: 'We wish to thank you for the service the Autocar trucks have rendered the farmers this summer, and trust they will be with us another year. It has been discouraging to them doubtless many times on account of the roads being in such condition, but we expect that the townships will improve them before another season, at which time we hope to see the trucks again.'"

The following is another case that has interested me particularly:

The Bush Line, of Wilmington, Del., runs freight steamers between Philadelphia and Wilmington. While this line has always maintained a collection and delivery service, it is only within the past few months that they have added motor trucks to their equipment. In talking to Mr. Bush the other day he said that the Autocars had not been working long enough to have set up actual figures of cost and profit, but, he said: "We are selling nothing more or less than service, and we have had a great many expressions of appreciation of the improved service that our motor trucks are rendering." Here is a case where Autocars are proving their efficiency on short-haul work, for, although none of the hauls is over three miles, the trucks are averaging 35 to 40 miles a day, as against 12 to 15 miles for a two-horse team.

In one sense, these trucks are working under ideal conditions, for they are owned and operated by the company that owns the piers, and every possible step is taken to make them as efficient as possible. The space of one side of the wharf is given to the trucks, and the freight is assorted with that in mind.

The business houses that are served by Autocars are saved a great deal of time, as the trucks cover a route in less than half the time that horses take. The Bush Line finds that on the item of overtime labor alone the Autocars are effecting a big saving. The trucks get to the pier in the evenings from a half to three-quarters of an hour earlier than the horse-drawn wagons, which means a great deal, as overtime is paid for all dock work after 6 o'clock.

There is no doubt but that the motor truck is a time-saver, and by that very reason alone it is a logical and sound step in the evolution of transportation.

Ship Stability—The Need for Investigation

BY A. T. WALL *

In spite of all the work that has been done in connection with the stability of ships, very little attempt is made to apply the results practically. It is true that in most shipyards nowadays the stability of a vessel is investigated in the design stages, but that in itself, useful and necessary as it is, is not sufficient.

Whatever precautions the designer may take, he can only allow for certain contingencies. After leaving the builder's hands a ship will be loaded up in a great many different ways, and it is impossible to predict all these conditions. Nearly every loading must be considered individually if anything like a true idea is to be formed of the stability of the vessel. In cases where similar, or nearly similar, cargoes are constantly carried it would be sufficient to determine the condition once, but in the majority of cases the load in a ship varies with every voyage.

At present the legal responsibility for the safety of a ship rests with the captain. Much legislation has been passed recently in regard to the building and running of merchant vessels, but neither the new laws nor the old ones, with one exception, make any mention of safe stability. The exception referred to has reference to the stability of passenger steamers. The British Board of Trade stipulates that the stability of any passenger steamer should be sufficient to render the vessel safe. It sometimes happens that the Board steps in and insists upon some additional stability being given to a ship by some means or another before granting the passenger certificate. No definition has ever been advanced by the Board of Trade as to what they consider to be sufficient stability, and in any case, as the stability they consider necessary can only take account of the ship in a general and not in a particular way, it amounts to the same thing as a good designer does for the ship, and the same criticism applies.

That accidents do happen to vessels through lack of stability needs no proof; numerous cases are on record where the facts have been established, and other instances come to mind in regard to vessels lost at sea without leaving any trace. Although the expression "safe stability" is used, it would be difficult to define it in exact terms. Metacentric height by itself is no criterion; a value which would make one ship safe might easily be dangerous for another vessel. Range of stability; that is, the angle through which a ship would roll (provided no water was shipped) without losing her righting moment, is equally no criterion when taken alone. Neither is the

* Naval architect, Messrs. Cammell, Laird & Co., Birkenhead.

maximum righting lever any more valuable. Even a knowledge of all these three quantities is not sufficient to form an opinion as to a vessel's safety. Absolute size of the ship, as well as the area of top hamper exposed to the wind, are factors which play important parts. Theoretical considerations alone cannot determine safe stability. In this matter, perhaps more than in any other, practice and theory must be combined. It is not suggested that naval architects do not know how all these qualities should be combined in a ship to make her quite safe, but it will generally be found that different experts would give different combinations for the same ship, in accordance with the dictates of their experience. This simply means that there is no standard for safe stability, and to set one up an enormous amount of careful investigation on ships in commission would have to be carried out.

At present ships go to sea without any exact knowledge of their stability conditions being available; in consequence, it is impossible to form any useful comparisons for future guidance. When a new ship is ready to sail some very elementary particulars of her stability may be given to the captain, often in such form as to be quite useless to him. They will generally be put away carefully and hardly referred to again. The captain then proceeds to form his own idea of his ship's stability. The vessel will sail on her first voyage, except in a few cases, with too ample stability, as the captain naturally does not wish to run any risks. From her behavior on the maiden voyage the captain forms his conclusions for future guidance, and ever afterwards the stability is practically fixed by rule of thumb.

Fortunately, the stability of a ship can vary within very wide limits without serious results. Too much stability is not good for a vessel; it unduly strains the structure and makes the ship uncomfortable in a seaway. If this excess of stability is obtained by the use of water ballast, then needless weight is carried about with the ship. On the other hand, if the stability is too small, the danger is too obvious to need detailed comment.

While rule-of-thumb methods are adopted to decide the stability of ships, many disadvantages will result, and sometimes with serious consequences. Numerous examples can be given showing how even an experienced captain may be deceived in regard to the condition of his vessel. Many of these come to light, but a far greater number probably go unrecorded. In one instance a ship was thought to be tender when loading, and water ballast was put in at the expense of cargo. At sea the vessel was found to be quite stiff, and the extra water ballast that had been taken in was pumped overboard, the ship still being amply stable. In another case a ship was behaving badly at sea, and more ballast was taken in, with the intention of rectifying the defect. This proceeding in itself was dangerous, but fortunately nothing serious happened. Actually, however, the ship became worse, and later it was found that her initial bad behavior was due to the fact that she was already too stiff. Perhaps a somewhat glaring instance of wrong procedure occurred with a small ship putting into a port on the East Coast some few years ago. The vessel had a list, and one lifeboat on the high side was filled with water, with the intention of remedying it. The ship then listed to that side; the opposite lifeboat was now filled with water, and shortly afterwards the vessel turned turtle. Quite recently the Dutch seagoing motor boat *Zeearend* was taking in general cargo at Rotterdam, and when the deck load was almost complete the vessel capsized. The capsizing of the *Eastland* is still fresh in memory, and illustrates a remarkable case of incompetency and ignorance on the part of those responsible for the safety of the ship. It was common knowledge that the vessel was tender and that water ballast had to be used to render her safe for the carriage of passengers. In spite of this, the ballast was omitted with results that are well known.

Such happenngs as these, and many more like them, would be next to impossible if some systematic efforts were made to provide those responsible with reliable information wherewith they could assess the stability of their vessels. It cannot be said that mercantile captains are not capable of appreciating the points involved, for in some cases—especially among the larger lines of passenger steamships—the captains are given information and know how to use it, and the results of their experiences are placed on record for use in future designs.

Quite apart from the disadvantages that arise from the lack of knowledge of the stability of a vessel, there are many advantages accruing from possession of all the facts. In the first place, no superfluous ballast need be carried about, giving a direct saving in coal consumption. At other times, with a ship nearly loaded down to her Plimsoll marks, it becomes necessary to decide whether ballast should be taken in to ensure sufficient stability, or if the remaining weight may be used for additional cargo. As often as not—from ignorance of the true position—the ballast is taken, although a little investigation might show the vessel to be amply safe. In entering shallow waters it is not uncommon for the captain to trim his vessel on to a level keel to avoid excessive draft aft. In such cases definite knowledge of the minimum water ballast required from stability considerations would considerably assist him. Again, in making a journey from one port to another it would often be useful to the captain if he knew exactly how much water ballast should be added to compensate for the consumption of coal, fresh water, etc., as the ship proceeded on her voyage.

It is sometimes argued that there is no time to make suitable inquiries into the stability of a ship when she is loading up hurriedly at a port, as time is money, but the expense involved by so doing would generally be far more than compensated for by the reduced amount of water ballast carried, as well as, in some cases, by the extra cargo that could be taken. When a ship is running light the circumstances are entirely changed. Water ballast, and as much of it as possible, must then be carried to increase the draft of water and give her a "grip" in the water, as well as to provide good propeller immersion.

As has already been remarked, a large amount of labor would be involved in setting up some definite standard of stability. Before the war it had been decided in Germany that a central bureau should be set up for the purpose of collecting information on the behavior of ships at sea, in addition to which calculations would be made so that the whole problem of stability might be placed on a proper basis.

A few contributions have been made to technical societies on the subject. In 1913 two papers were read before the Schiffbauteschnische Gesellschaft, one by Herr Ludwig Benjamin and the other by Dr. Carl Commentz. The former considered that minimum values should be fixed for the righting levers at two angles of inclination of the ship, while the latter urged the necessity for careful investigation of stability with the object of recommending minimum stability curves for different ships. A year later a paper was contributed to the Institution of Naval Architects by the writer, giving a method for the comparison of stability between different ships or the same ships under different conditions of loading.

These papers, however, only touch the fringe of the

subject. Joint action on a large scale is necessary if results of any value are to be obtained. It is not sufficient to confine attention to a ship in still water; rolling in a seaway, wind pressures, virtual loss of weight of the ship in the trough of a sea, shifting of weights and shipping of water are some of the things that must be allowed for. The behavior of ships at sea whose stability conditions are well known needs to be carefully noted, and when good results have been obtained they can be used as a working basis. Incidentally, the stability of ships when damaged could be given attention—a study in itself.

The present-day knowledge of stability is quite sufficient to ensure that no vessel need put to sea with any uncertainty as to the sufficiency of her stability. The dangers arise from lack of application of the known results. The responsibility for safe loading should not, on any account, be taken from the captain, but he could be enormously assisted in carrying out his duties in this direction if the necessary steps were taken. In the larger field of what might be termed the determination of minimum safe stability there can be no doubt, in view of the resulting advantages, that the matter will sooner or later be taken up in earnest, with far-reaching results for the benefit of shipowners and seafaring men.

United States Naval Experimental Wind Tunnel at Washington Navy Yard

Description of the Largest Wind Tunnel in the World—Wind Speeds up to 75 Miles an Hour Available for Experiments

BY NAVAL CONSTRUCTOR WILLIAM M'ENTEE, U. S. N.

The large experimental wind tunnel which the Navy Department has established in the Washington Navy Yard at the Experimental Model Basin, where warship models are tested, has now been in operation about a year. The tunnel is the largest in the world, having a section 8 feet square at the point where the models are placed for testing. In addition to the advantage gained by the size, it is possible with the 500-horsepower, motor-driven fan to get wind speeds up to 75 miles an hour, which permit experiments being made at real flying speeds.

The tunnel consists of a closed circuit shaped like the link of a chain, as shown in Fig. 1. The 500-horsepower top horizontal discharge fan of the corrugated paddle type, with an inlet diameter of 11 feet 2 inches, and a discharge duct 7 feet 6 inches by 9 feet, is placed at one end of the link. At the other end, where the air straightens

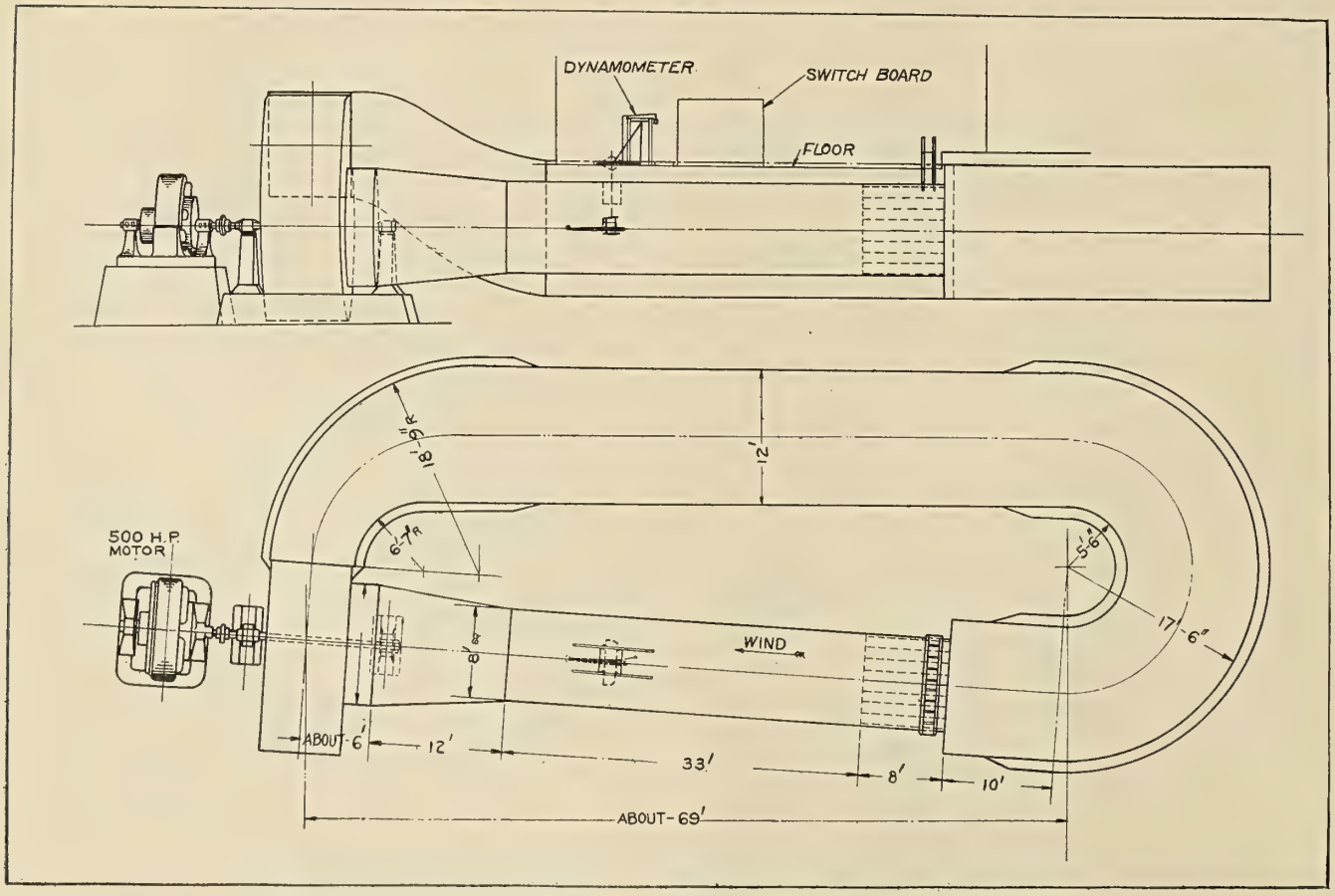


Fig. 1.—Diagram Showing General Arrangement of Naval Experimental Wind Tunnel

out before flowing through the experimental chamber, are the baffles, which are necessary to remove the eddies and to control the uniformity of the speed. These baffles consist of 64 cells, each 1 foot square and 8 feet long. Each cell is provided with its own damper, so that the velocity of the air in any one section may be controlled. At the experimental chamber in the vicinity where aeroplane wings or models are tested the maximum variation from uniform flow is about 2 percent.

The tunnel is built of wood, with frames spaced about 3 feet on centers placed outside and sheathed on the inside, with 7/8-inch tongued and grooved sheathing laid in two thicknesses in the direction of the air current, and with building paper placed between the two layers. The necessary curvature is obtained by bending the sheathing, the whole of which is blind nailed.

The fan is driven by a 250-volt, 500-horsepower, direct-current motor, arranged for operation on the Ward-Leonard system. The motor also has auxiliary field control, so that any desired speed up to about 200 revolutions per minute, which corresponds to a wind speed of 75 miles an hour, may be obtained. At the discharge side of the fan are located 12 pitot tubes which lead to an integrating manometer which gives the average velocity of discharge. This velocity has been calibrated against the velocity obtained at the section in the experimental chamber, where the aeroplane or other model is placed, so that any desired velocity may be obtained at that point with precision without having any pitot tubes or obstructions other than the model being tested. In other words, by calibration the velocity of discharge may be found, and this bears a certain constant ratio to the velocity at the experimental section.

The velocities were determined by pitot tubes which were checked with those used in the Aerodynamical Laboratory of the Massachusetts Institute of Technology, and in the National Physical Laboratory in England.

Among recent investigations of interest made at the wind tunnel was the determination of the coefficient of air friction for various aeroplane and balloon fabrics. Tests have been made on the new dirigible building for the Navy Department, and on models of naval aeroplanes both building and projected. A number of tests have also been made for private concerns. In carrying out experiments for private parties, the same practice is followed as in the case of tests of ship models; that is, the actual cost of doing the work is charged in each case. On account of the large size of the tunnel it is possible to test comparatively large models of aeroplanes with widths up to 36 inches.

Fig. 2 is a photograph showing the arrangement of the model of an aeroplane when being tested. The model is carried by a steel spindle which extends up through the top of the tunnel to the weighing balance, which is placed overhead. For about two-thirds of the length of the tunnel the spindle is covered by a mask of stream line form. This mask is secured to the ceiling of the tunnel and reduces the force acting on the spindle itself, and thus the spindle correction. The weighing balance consists of a weighing scale on the platform principle having three axes, two of them at the same horizontal line 61 inches apart, and the third vertically over one of the first, 48 inches above it. When a model is set at a given angle, the moments acting about each of these axes are measured by weighing them on the scale. With these data it is possible to compute horizontal and vertical components of the force acting on the model; that is, the drift and lift, and also to compute the line of application of the force. Tests are usually made at speeds of 40 miles an hour. At this speed and at the angle of least resistance an ordinary aeroplane

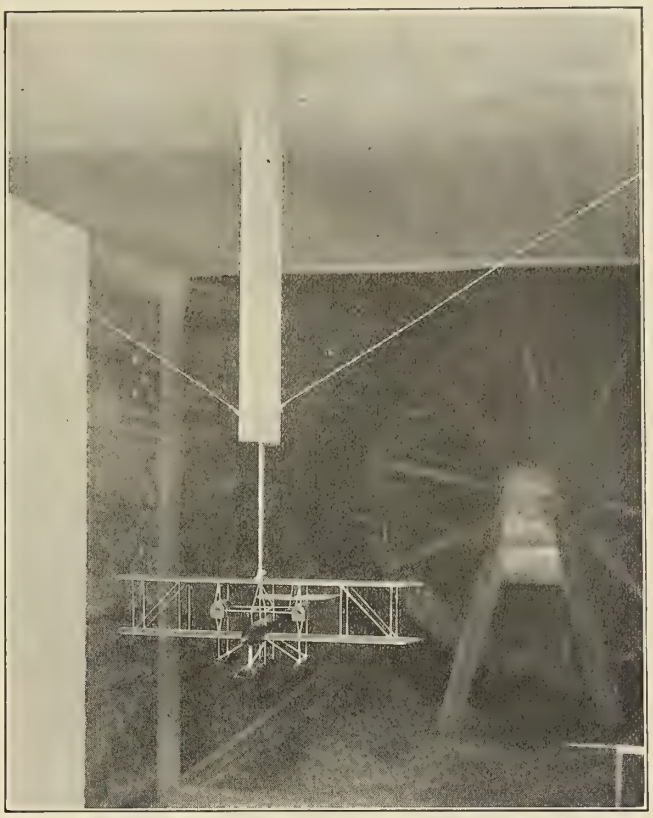


Fig. 2.—Testing an Aeroplane Model

wing model has a horizontal resistance of something less than one-tenth of a pound. It is therefore necessary that the balance should be capable of weighing a force with accuracy to about 2/1,000ths of a pound.

The large size of the tunnel makes it possible to test full-size radiators for aeroplane motors, and comparative tests have recently been made on several types both as to air resistance and cooling capacity.

The wind tunnel has also been used for certain other tests which are not directly connected with aeronautics, such, for example, as the determination of the influence of form and dimensions on the size of ventilating cowls for use on ships. These tests have shown that it is not necessary to exceed certain dimensions which are less than have heretofore been used in many cases. It is proposed shortly to obtain the wind resistance of a large battleship, this being an element of a ship's resistance which has not previously been accurately measured.

In conjunction with the wind tunnel, the model basin is used for determining the best form of floats for hydroplanes, and there will no doubt be many other cases where the data derived from tests made in the water may be applied directly to aeroplane design.

Vessels Built in United States.—The Bureau of Navigation, Department of Commerce, reports 32 sailing, steam, gas and unrigged vessels of 20,307 gross tons built in the United States and officially numbered during December, 1915. The largest steel steamers included in these figures are: *Floridian,* of 6,930 gross tons, built by the Maryland Steel Company, Sparrows Point, Md., and owned by the American-Hawaiian Steamship Company; the *Munamar,* of 3,477 tons, for the Munson Line, also built by the Maryland Steel Company; and the *Bylayl,* of 3,289 tons, built by the New York Shipbuilding Company, Camden, N. J., and owned by the Pocahontas Navigation Company.

Propeller Design for Turbine-Driven Ships

Methods of Determining the Principal Dimensions of Three-Bladed, High Tension, Bronze Propellers for Turbine-Driven Vessels

BY N. A. GRAVESON *

The propellers for turbine-driven vessels, owing to the high number of revolutions at which they run, have brought forward a different design of propeller to that generally used on reciprocating driven vessels. The latter follow more or less the elliptical form advocated by the late Dr. Froude—i. e., long blade and relatively narrow width of blade, but the turbine-driven propeller running at a high speed and having to obtain a large blade surface, so that the pressure per square inch on the blade surface can be kept between certain limits, assumes a disk-like and sometimes a pear-shaped form of blade. The ratio of projected blade surface to the disk area of the propeller advocated by Dr. Froude was .2 to .26, and in later years destroyer propellers rose to about .35, while turbine-driven propellers can be said to start at .4 and rise to .8, the latter figure being for light, fast vessels.

The principal dimensions of propellers depend upon the following:

$S.\ H.\ P.$ = shaft horsepower developed in the shaft and delivered to the screw.
$E.\ T.\ P.$ = effective thrust power = $S.\ H.\ P.$ × suitable factor.
V = the speed of the ship in knots.
S = apparent slip of the screw.
T_r = total resistance of the ship.
R = revolutions of the screw per minute.
U = speed of the wake in knots.

And also upon the type and shape of the vessel, especially the stern.

Revolutions

The revolutions of turbine-driven propellers range through the following: 200 and 300 revolutions per minute for large vessels with full lines, to 800 revolutions per minute for fast, fine-lined vessels.

Apparent Slip

The apparent slip is the following:

$$\text{Apparent slip} = \frac{\text{speed of screw} - \text{speed of ship}}{\text{speed of screw}}$$

all being in the same units. The apparent slip depends upon the revolutions, the speed and shape of the vessel, the blade surface, and the diameter of the propeller.

In dealing with the apparent slip mention may be made of the actual slip. The propeller works in the wake stream of the vessel, and this stream of water follows the forward motion of the vessel, being greatest in ships with full stern lines. The actual slip is given by the following:

$$\text{Actual slip} = \frac{\text{speed of screw} - (\text{speed of ship} - \text{speed of wake})}{\text{speed of screw}}$$

In practice it is usual to work off the apparent slip, as the actual slip can only be arrived at when the velocity of the wake is known—and it is difficult to arrive at this.

The apparent slip of turbine-driven screws, between 200 and 800 revolutions per minute, is given by the following empirical formula:

Let R = revolutions per minute of screw and S = apparent slip.
Then $S = .0206\ R + 12$.

There will be deviations from the slip given by the above, owing to tides, etc., but it will be found to hold good for general purposes.

Diameter

To get the diameter of a propeller, it is necessary to be able to deduce the effective thrust. This is a long and tedious process to do theoretically and involves such a large number of factors that the practical man has not the time to spare, and it is, therefore, best to use an empirical method which has given good results in practice.

This consists of employing suitable coefficients to suit various types of vessels, and this method was advocated by Mr. Speakman in a paper before the Institution of Engineers and Shipbuilders in Scotland. C, the coefficient, depends upon the pressure per square inch on the blade surface and the ratio of the projected blade surface to the disk area of the propeller, and is applied as follows:

$$C = \frac{\sqrt{\text{effective thrust in pounds}}}{\text{diameter of propeller in feet}}$$

The effective thrust is the power required to propel the vessel and is equal to the shaft horsepower multiplied by a coefficient varying from .55 to .52.

The effective thrust in pounds is given by the following formula:

$$E.\ T. = \frac{S.\ H.\ P. \text{ on one shaft} \times 33{,}000 \times .52 \text{ or } .55}{\text{speed of vessel in knots} \times 101.3}$$

The table below gives the value of C and ratio of E. T. P. to S. H. P. for various vessels:

Type of Vessel	Speed of Vessel	Approximate Revolutions of Turbines	E.T.P. / S.H.P.	Apparent Slip, Percent	C
Large ocean mail steamer	24 to 25	190 to 200	.53	16.5	23
Intermediate ocean mail steamer	21 to 23	330	.55	18.8	24.6
Cross channel steamers	24	500 to 550	.53	20.5	25
Fine-lined fast vessel	28	750	.52	27.5	30

For vessels coming in between the above types, a value of C can be chosen to suit by using a little judgment.

Surface

The surface of a propeller depends on the pressure per square inch allowed on the projected blade surface, but as the projected blade surface can only be found after the developed surface it is best to use the pressure per square inch on the developed surface. This lies between 9 and 12 pounds per square inch, neglecting that due to the depth of immersion of the screw. It is highest for fast vessels and lowest for the slower type of vessels and, as the revolutions are fixed by the type of vessel, it will be best to consider the pressure per square inch as a factor of the revolutions, as in the following:

Pressure per square inch on developed surface = $.00563\ R + 7.5$.

To show method of arriving at the principal dimensions of a propeller, the following example is taken:

Propeller For a Cross-Channel Steamer

Number of shafts = 3. $S.\ H.\ P.$ = 9,500 on three shafts.
Revolutions per minute of turbines = 520.
Speed of vessel = 23 knots. $\frac{E.\ T.\ P.}{S.\ H.\ P.} = .53$.

* Marshall Scholar.

C, coefficient for diameter of propeller, = 25.

Diameter of Propeller in Feet.—

$$\text{Effective thrust} = \frac{\text{S. H. P. on one shaft} \times \frac{\text{E. T. P.}}{\text{S. H. P.}}}{\text{speed of vessel in feet per minute}}$$

$$\text{Effective thrust} = \frac{9{,}500 \times 33{,}000 \times .53}{3 \times 23 \times 101.3} = 23{,}780 \text{ pounds.}$$

$$\text{Diameter} = \frac{\sqrt{23{,}780}}{25} = \frac{154.2}{25} = 6.17, \text{ say 6 feet 2 inches diameter.}$$

Pitch.—The apparent slip of propeller is given by the formula $.0206\,R + 12 = .0206 \times 520 + 12 = 22.712$ percent.

Required speed of screw to drive vessel at 23 knots

$$= 23 + \frac{22.7}{100} \times 23 = 23 + 5.22 = 28.22 \text{ knots.}$$

$$\text{Pitch} = \frac{28.22 \times 101.3}{520} = 5.49 \text{ feet, say 5 feet 6 inches pitch.}$$

Surface.—Pressure per square inch on developed surface $= .00563\,R + 7.5 = 10.42$ pounds per square inch.

$$\text{Area in square inches} = \frac{\text{effective thrust in pounds on one propeller}}{\text{10.42 pounds per square inch developed surface}}$$

$$= \frac{23{,}780}{10.4 \times 144} = 16 \text{ square feet, nearly.}$$

DIMENSIONS

Diameter—6 feet 2 inches.
Pitch—5 feet 6 inches.
Developed Surface—16 square feet.

It is a good policy when fixing up the propellers for fast vessels to keep the pitch on the small side and the surface and diameter on the large side, as this will allow for modifications after the trial, and, if necessary, the diameter can be slightly reduced, which will allow the turbines to be run a few revolutions faster to suit a less thrust at a higher velocity.

Summary of the foregoing notes:

Type of Vessel	Speed of Vessel, Knots	Approximate Revolutions of Turbines	E.T.P. / S.H.P.	Apparent Slip of Propeller, Per-cent	Pressure per sq. in. on Developed Surface, Pounds	Ratio of Developed Surface to Disk Area	C Coefficient for Diameter
Large ocean mail steamer	24 to 25	190 to 200	.53	16.5	8.75	....	23
Intermed'te ocean mail steamer	21 to 23	330	.55	18.8	9.35	.55	24.6
Cross channel steamer	24	500 to 550	.53	20.5	10.3	.535	25
Fine-lined fast vessel	28	750	.52	27.5	11.72	.73	30

$$\text{Diameter of propeller in feet} = \frac{\sqrt{\text{Effective thrust in pounds.}}}{C}$$

Apparent slip of propeller, percent, $= .0206\,R + 12$.
Pressure per square inch of developed surface $= .00563\,R + 7.5$.

Constructional Details

The length of the propeller boss is fixed by the compressive stress per square inch allowed on the sides of the keyway in the boss, due to the tangential force of the turning moment on the propeller shaft, and to arrive at this stress the diameter of the shaft and the size of the propeller key must be known.

A suitable example will illustrate the method used, and, for preference, the propeller for the channel steamer will be taken.

Size of Tail-End Shaft

Let S = diameter of shaft in inches.
$I.H.P.$ = estimated maximum indicated horsepower transmitted through shaft.
R = revolutions per minute of shaft.
f = 60.3 for tunnel shafts.
f = 82.8 for propeller shafts.

Then the Board of Trade's formula for turbine shafts is

$$S = \sqrt[3]{\frac{I.H.P. \times f}{R}}$$

Now the indicated horsepower of a turbine is generally taken to be the shaft horsepower divided by .9. Then the indicated horsepower per shaft for a channel steamer with three shafts is

$$\frac{9{,}500}{3 \times .9} = 3{,}518\ I.H.P.$$

and the size of the propeller shaft is

$$\sqrt[3]{\frac{3{,}518 \times 82.8}{520}} = 8\tfrac{1}{4} \text{ inches, nearly.}$$

Size of Key

There are generally two keys in the propeller shaft, as shown in Fig. 2. To get the shear stress in the keys, first

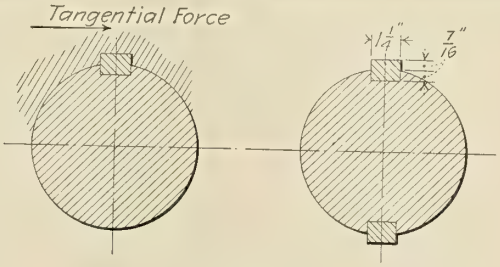


Fig. 1 Fig. 2

take the length of taper on propeller shaft. Rule for taper = 2.1 × diameter of propeller shaft in inches. Then, length of taper on shaft ends = 2.1 × 8¼ = 17½ inches. Now the keys are made ½ inch less at each end than the length of the taper, and the length of the key is 16½ inches long. From this the breadth of the key can now be arrived at by allowing a shear stress of 2,500 pounds per square inch through the cross section of the key where it intersects the shaft diameter. The stress, 2,500 pounds, is that generally used for medium type propellers, and it is taken up to 3,000 pounds per square inch for fast, high speed propellers, and for low speed propellers, 2,000 pounds shear stress is allowed. If a higher stress than 3,000 pounds per square inch is allowed through the cross section of the key, the tangential pressure on the key becomes abnormal and would necessitate a larger standout of key into the propeller boss, which only weakens the boss, so it is better to keep the shear stress in the keys between the figures given above.

Key to Channel Steamer Propeller

The shaft is 8¼ inches diameter, the taper on shaft 17½ inches long, and this taper is generally 1 in 12. Therefore, the diameter of shaft at centerline of propeller is

$$8\tfrac{1}{4} - \frac{8\tfrac{1}{4}}{12} = 7\tfrac{1}{2} \text{ inches.}$$

Tangential force on shaft at this diameter

$$= \frac{\text{S. H. P. on one shaft} \times 33{,}000}{\frac{7\frac{1}{2}}{12}\pi \times \text{revolutions}} = \frac{9{,}500 \times 33{,}000}{3 \times \frac{7.5}{12}\pi \times 520}$$

$$= 102{,}100 \text{ pounds.}$$

$$\text{Breadth of keys} = \frac{\text{tangential force}}{\text{length of key} \times 2 \times \text{shear stress (because there are two keys)}}$$

$$= \frac{102{,}100}{2{,}500 \times 16\frac{1}{2} \times 2} = 1\frac{1}{4} \text{ inches, nearly.}$$

$$\text{Depth of standout of key into propeller loss} = \frac{\text{tangential force on one key side}}{\text{area of side of key in way of propeller boss} \times \text{compressive stress allowed}}$$

The compressive stress allowed is as follows: 9,000 pounds per square inch for light, fast running propellers;

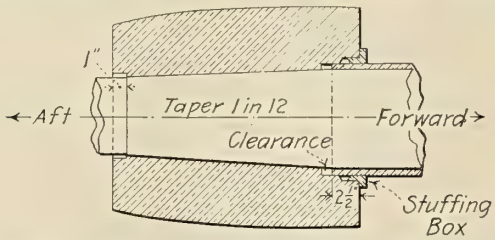


Fig. 3.—Sketch Showing Allowance on Forward and After Ends of Propeller Boss

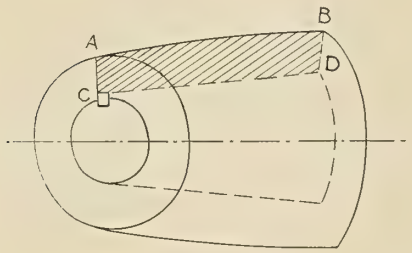


Fig. 4

6,000 pounds per square inch for propellers where weight is not considered too much.

Allowing a stress of 7,100 pounds on side of key, the standout or height of key becomes

$$\frac{102{,}100}{16.5 \times 2. \times 7{,}100} = 7/16 \text{ inch.}$$

Diameter of Propeller Boss

The diameter of the boss should be about twice the diameter of the propeller shaft, provided there is sufficient metal through the cross section of the boss in way of the key to prevent it bursting or cracking owing to the thrust on the taper of propeller shaft.

Length of Boss

The length of the propeller boss is the length of the shaft end plus 1 inch allowed on the aft end to enable the boss to be bored out to a slightly larger taper if necessary when fitting a new tail end shaft, and also 2½ inches to be allowed at forward end of taper to fit a small stuffing box and to allow clearance at end of shaft taper for tightening up purposes.

Propellers designed for high speed boats, where weight is cut down to the utmost limit, have usually a small diameter propeller boss, and this is fixed up by the bursting stress allowed through the cross section of the boss. Thus bursting stress may be allowed to go as high as 9,500 pounds per square inch, but it is not advisable to do this in ordinary practice.

Bursting stress in boss through area $A\ B\ C\ D$ (Fig. 4):

$$= \frac{\text{total thrust} \times \text{reciprocal of taper}}{\text{area } A\ B\ C\ D}$$

Example.—Size of boss for cross channel steamer propeller, allowing a bursting stress of 5,000 pounds per square of cross section of boss. Length of boss = length of taper + 3½ inches = 21 inches.

$$\text{Total thrust} = \frac{S.\ H.\ P. \text{ on one shaft} \times 33{,}000}{\text{speed in knots} \times 101.3}.$$

$$\text{Then thickness of metal} = \frac{S.\ H.\ P. \times 33{,}000 \times 12}{\text{length of boss} \times V \times 101.3 \times 5{,}000}$$

$$= \frac{9{,}500 \times 33{,}000 \times 12}{3 \times 21 \times 23 \times 101.3 \times 5{,}000} = 5\frac{1}{4} \text{ inches, nearly.}$$

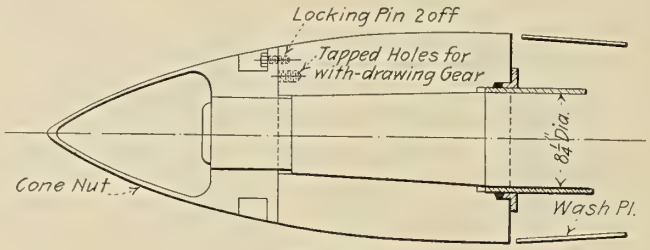


Fig. 5

Shape of Boss

A large cone nut is fitted to the propeller to overcome the frictional loss due to the propeller boss. This cone nut should be the same length as the propeller boss and forms with the boss a true stream line shape (see Fig. 5).

If the diameter of the propeller boss is different from that of the strut bearing, the wash or eddy plates must be made to suit.

In choosing the diameter of the propeller boss, it should be noted that a large diameter boss is more efficient than a small one, as the thrust near the boss is small; and not only that, a large boss allows the lock pins which hold cone nut from turning and the holes for withdrawing gear to be kept away from small end of taper, and also allows of a large size of withdrawing screw being fitted.

A short stuffing box is fitted to the forward end of the boss to prevent water creeping along the shaft taper, and this stuffing box is packed with an india rubber washer.

Cone Nut

The cone nut for the propeller is screwed opposite hand to the propeller and has two flats cut on it to engage the spanner for tightening up. Care should be taken when designing this nut to have sufficient length of thread to take propeller thrust when going astern and the length of nut should be equal to the small end of the shaft taper at the very least. The thickness of metal in the thin stream line portion of the cone nut should be ⅜ inch for small propellers and ⅝ inch for large ones.

Gland at Forward End of Boss

This gland is to be made in halves and is fitted with ¾-inch, square-necked studs about 3½ inches pitch.

Thickness of Blade Root

There are two forces acting on the root of the propeller blade, namely, the bending moments, due to the thrust and the tangential force of the turning moment of the propeller shaft, also the centrifugal force due to the weight of the propeller blade and the revolutions. These bending moments are caused by the resolved forces of the thrust and tangential forces acting at a radius equal to the distance from the center of the shaft to the center of thrust minus the radius of the propeller boss. The center of thrust for average blades is taken to be ⅝ radius of

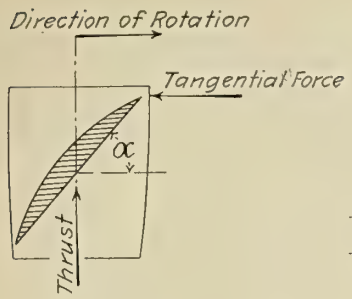


Fig. 6.—Forces Acting on Root Section of Blade

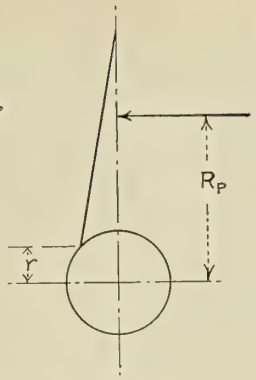


Fig. 7

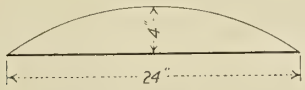


Fig. 8

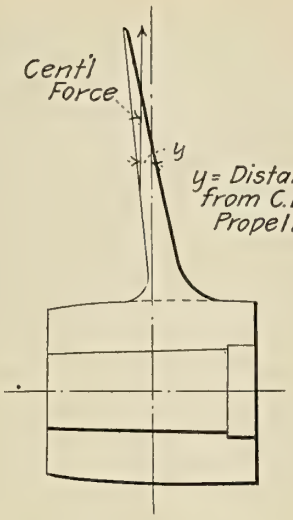


Fig. 9.—Sketch Showing Effect of Centrifugal Force on Root Section of Set Back Blade.

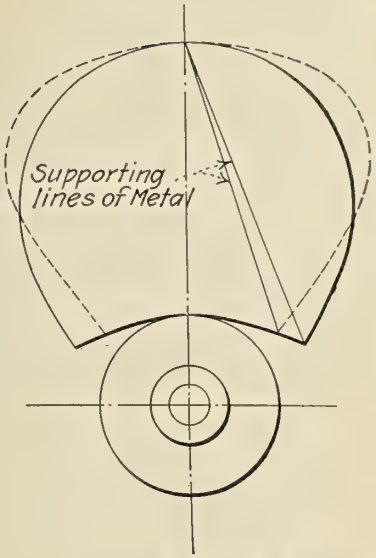


Fig. 10.—Sketch Showing Blade Shapes

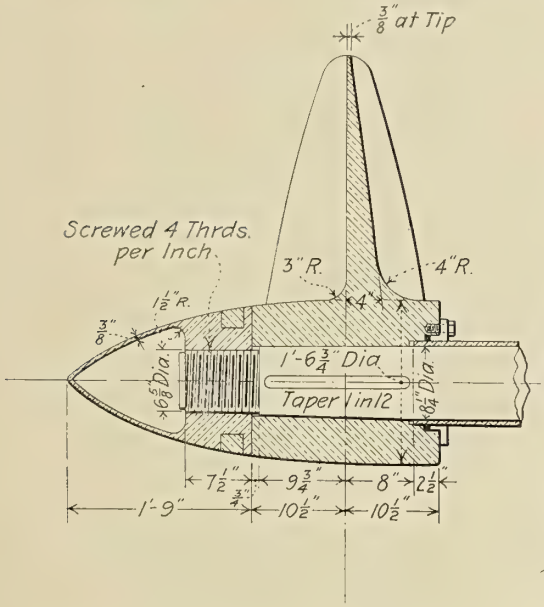
the propeller from the centerline of the shaft, and the resolved forces become (see Fig. 6)

Resolved force of thrust = Total thrust in pounds × cosα
Resolved tangential force = Tangential force × sinα

$$\text{Total thrust on one blade} = \frac{S.H.P. \text{ on one shaft} \times 33{,}000}{\text{number blades} \times V \times 101.3}.$$

$$\frac{S.H.P. \text{ on one shaft} \times 33{,}000}{2\pi \dfrac{\frac{5}{8} R_P}{12} \times \text{revolutions}}$$

and this acts at a leverage of ⅝ radius of propeller minus



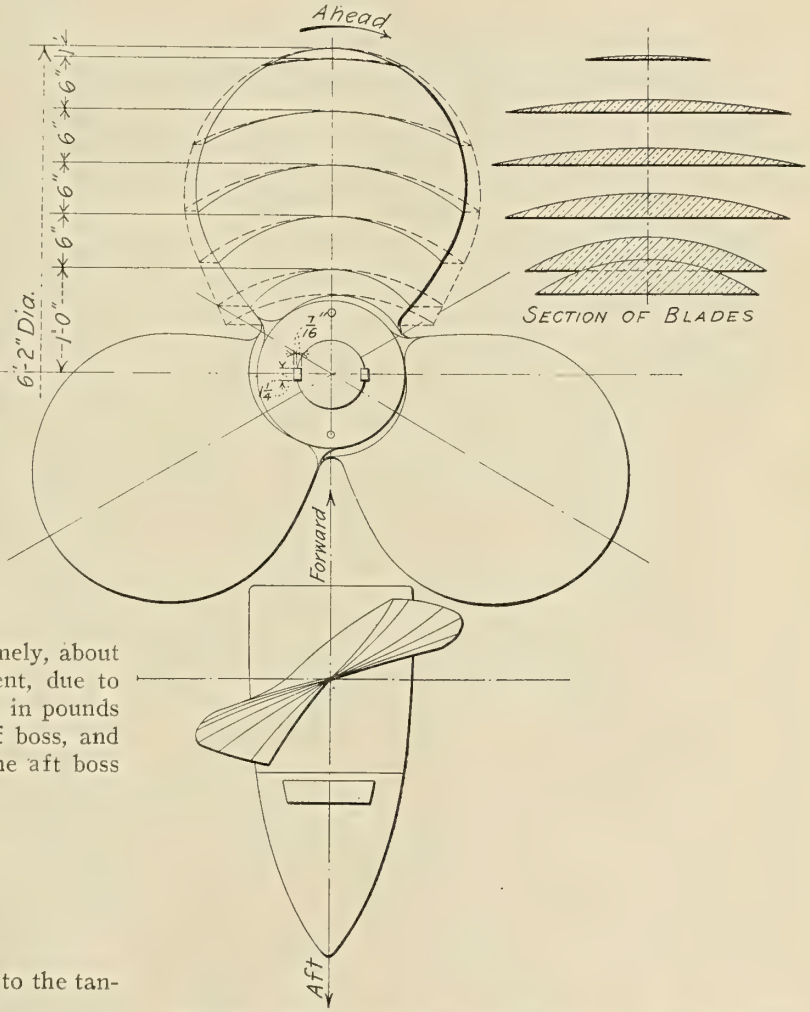


Fig. 11.—Right Hand Screw Propeller for Channel Steamer. Diameter, 6 Feet 2 Inches; Pitch, 5 Feet 6 Inches; Developed Area, 16 Square Feet; Projected Area, 14.2 Square Feet

This force will act at the center of thrust, namely, about ⅝ radius of propeller, and the bending moment, due to the resolved force of the thrust = Total thrust in pounds cosα × ⅝ radius of propeller minus radius of boss, and the stress in the blade root in tension along the aft boss of the blade will be

$$\frac{\text{thrust} \cos\alpha \times \frac{5}{8} R_P - \text{radius of boss}}{\text{modulus of section}}$$

$$\text{Modulus of section} = \frac{Bh^2}{9}$$

Again to this stress we must add the one due to the tangential force of the turning moment.

Tangential force due to turning moment of shaft at ⅝ radius of propeller equals

radius of boss. To simplify these stresses the following will be used:

Let $S.H.P._1$ = $S.H.P.$ on one shaft.
T_1 = total thrust divided by number of propeller blades.
R_p = radius of propeller in inches.
r = radius of propeller boss in inches.
V = speed of vessel in knots.
R = revolutions per minute of screw.
α = angle in degrees that blade root makes with *athwartship* centerline.
h = thickness of blade root in inches.
B = width of blade root in inches.
f_c = tensional stress in blade root due to centrifugal force (not to exceed *3,000* pounds per square inch).
f_t = tensional stress in blade root due to thrust and tangential force. This stress is kept between *5,000* and *8,000* pounds per square inch so as not to exceed a safe stress when the stress due to the centrifugal force is added to it.

The easiest method to be used to get the thickness at the root of the blade is to reduce the bending moments to a known force and then equate this to the modulus of section of the blade root times stress to be allowed, then

$$h = \sqrt[2]{\frac{9(\text{thrust } B.M. + \text{tangential } B.M.)}{B \times f_t}}$$

When h is found, try out the stress due to the centrifugal force on the root section by the following:

$$f_c = \frac{.00034\, W \frac{R_p}{12} R^2}{\frac{2}{3} Bh}$$

where W = weight of one propeller blade in pounds, and if this stress, together with the other stress due to the bending moments, exceeds the safe stress allowed, increase h and again try out stress. It is best to use the method given above, as combining the stresses and trying to get the thickness of the root by a shorter method lead to confusion of the terms involved.

Example.—Thickness of blade root for channel steamer propeller. Breadth of blade = 24 inches.

$$T_1 = \frac{9{,}500 \times 33{,}000}{9 \times 23 \times 101.3} = 14{,}900 \text{ pounds.}$$

α = 50 degrees. Resultant force = 14,900 Cosα = 9,550 pounds. Tangential force at $\frac{5}{8} R_p$

$$= \frac{9{,}500 \times 33{,}000}{3 \times 2\pi \times 1.93 \times 520} = 16{,}600 \text{ pounds.}$$

16,600 sin. α = 12,800. Add these two together

9,550

= 22,300 pounds, and this acts at
radius = $R_p - r = 23.2 - 9.375 = 13.825$ inches.
$B.M.$ = 22,300 × 13.825 = 308,100 inch pounds.

Allow a stress of 7,000 pounds per square inch in tension along aft face of root and

$$h = \sqrt[2]{\frac{9 \times 308{,}100}{24 \times 7{,}000}} = \sqrt{16\tfrac{1}{2}} = 4.06, \text{ say 4 inches.}$$

Now try out the centrifugal force acting on this root section (Fig. 8) with weight of one blade, say 400 pounds

$$= \frac{.00034 \times 400 \times 1.93 \times 520 \times 520}{2/3 \times 24 \times 4}$$

= 1,100 pounds per square inch.

Then total stress at root section = 1,100 + 7,000 = 8,100 pounds per square inch. The tension at the root is only dealt with, as high tension bronze is weaker in tension than compression, the ultimate strengths being tension, 26 to 30 tons, compression, 60-70 tons per square inch.

Set Back of Blade

The propeller blades of turbine vessels should not be set back as in reciprocating practice, because this greatly increases the bending moment on the blade root owing to the centrifugal force of the blade acting outside the root section. (See Fig. 9.) A small set back on a very fast running propeller will raise the root stress by as much as 2,000 to 3,000 per square inch.

Thickness of Blade Tips

The thicknesses of blade tips should be as follows: 5-foot diameter propeller, 5/16 inch at tip; 12-foot diameter propeller, ¾ inch at tip.

The propeller blades should be finely polished to reduce skin friction, and where the propeller blade joins the boss a generous radius should be allowed.

Shape of Blade

The blade should be made as near as possible to a disk-like shape (see Fig. 10), as this shape is found to give very good results in practice. Pear-shaped blades, shown by dotted lines in Fig. 10, cause undue vibration, as the outer edges of the blades are too far from the line of support, which is a line drawn from the top of the blade to the root edge.

How Our Navigation Laws Compare with Others

The navigation laws of the United States have been a subject of spirited controversy of late. It has been stated that the navigation laws of the United States are inadequate and that they impose undue hardships on our merchant marine. On account of the lack of published information on the subject, there has been no direct means of determining whether or not these statements are correct. In order to set forth clearly and concisely the exact facts in reference to the navigation laws of the principal maritime countries of the world, the Bureau of Foreign and Domestic Commerce, Department of Commerce, will issue, within the next two or three weeks, an exhaustive report on the principal features of the navigation laws of Great Britain, Germany, Norway, France, Japan, and the United States. No such work has ever before been attempted in the United States, and it will be invaluable to those who are studying the problem of our shipping and merchant marine.

This publication will present the principal features of the navigation laws of the different countries in such a way to show just how our laws differ from those of other maritime countries. The abstracts of the laws were obtained direct from the foreign maritime laws and have been submitted to competent officials of the various countries for criticism and revision. No conclusions are to be drawn and no attempt is to be made to enter one side or the other of the controversy that has been raging for months past with reference to our navigation laws.

Largest Plate Mill in the World.—The Lukens Iron & Steel Company, Coatesville, Pa., has placed a contract with the United Engineering & Foundry Company, Pittsburg, Pa., for the construction of a four-high plate mill of the type used for rolling armor-plate at the plant of the Carnegie Steel Company, the length of the rolls of which will be from 200 to 204 inches, and which will be capable of rolling plates about 16 feet in width. The largest plate mill in the world at present is located at Witkowitz, Hungary, and has rolls 178 inches long.

Calculations for Ship's Forms*

Formulae, Deduced from Model Basin Experiments, to Determine Dimensions of Propellers for Best Efficiencies

BY REAR ADMIRAL D. W. TAYLOR†

Since screw propellers came into common use, some seventy-five years ago, there have been innumerable theories of their action, and the screw propeller is still a favorite subject for the inventor. It is only within the last ten or fifteen years, however, since fairly extensive model-propeller investigations have been made public, that the fog—if I may so express it—surrounding propellers has begun to clear away. Even yet model experiments with propellers are not so generally accepted as those for ships. One outstanding result, however, of model-propeller investigations during the last few years is a demonstration of the fact that in former theories of propeller action an important factor has usually been overlooked. The vast majority of propellers have sections of ogival type, with a straight driving face and the back an arc of a circle, or very close to that curve. In dealing with such propellers, the only factors usually considered have been diameter, pitch and blade area—the pitch being the uniform or average pitch of the driving face. Now, the diameter is really the primary factor affecting the action of a propeller. The pitch of the face is a convenient but purely conventional measure. We know, as a matter of fact, that an ordinary propeller, as described above, will develop a very material thrust when its slip, based upon the pitch of its face, is zero. In other words, with such a propeller the virtual pitch is always greater than the nominal or conventional pitch. The factor which has been ignored in this connection is the thickness of the blade, or, what amounts to the same thing, the shape of the back of the blade. The back of the blade has a very powerful influence on propeller action, and for the ordinary type of propeller described above, its shape is dependent upon the thickness of the blade. The amount of this influence was not at all realized until it was investigated by model experiments. For instance, published results of experiments at the U. S. Model Basin‡ show that two model propellers, each 16 inches (0.407 m.) in diameter, 19.2 inches (0.487 m.) face pitch, with a ratio of projected area to disk area of 0.3023, but differing in blade thickness, one being abnormally thin and the other abnormally thick, showed radically different results. Thus, at 5 knots speed of advance and 20 percent slip, the thin blade showed a thrust of 35 pounds (15.9 kg.), whereas the thick blade showed a thrust of 51 pounds (23.1 kg.), an increase of nearly 50 percent.

Blade area or surface is very unsatisfactory as a primary variable. This is essentially for the reason that the effect of variation of blade area depends upon how it is varied. In the case of a given propeller, if we increase blade area by increasing diameter, without materially changing the width of the blade, the effect is great. If we make the same increase of blade area by increasing the width of the blade, without increasing diameter, the effect is comparatively small; and if, as is apt to be the case in such instances, the increase is accompanied by a reduction of thickness, we may get no effect at all. For this reason, in any theory of propeller action, it is desirable to bring in the blade area in some manner which will make its effect always similar. Perhaps the most convenient method of doing this is to use, not the blade area, but the ratio of maximum or mean width of blade to the diameter.

In this way, any change of blade area due to change of diameter is taken up by the diameter factor, and the change of area due to change of width is taken up by the width coefficient, or, as it is conveniently called, the mean-width ratio. In any reliable propeller theory, then, we must take account of the diameter and mean width of the blade, and the real pitch. In practice, the real or virtual pitch depends partly upon the nominal pitch of the face, and partly upon the blade thickness. It is convenient to separate these factors and use as variables the nominal pitch and what I have called the "blade thickness fraction," namely, the ratio between the root thickness of the blade at the axis (obtained by extending the straight lines of face and back to the shaft center), and the diameter of the propeller. If we adopt these variables, it is possible, by semi-empiric methods, to express the results of any systematic series of propeller experiments by formulæ. Of course, the results of such experiments must first be expressed graphically, and, for practical purposes, complete graphic expressions are as good as formulæ; but most people seem to prefer to work out a result by means of a formula rather than from graphic data.

Some years ago* there were published the results of experiments, at the U. S. Model Basin, with 120 different model propellers, varying in pitch and mean-width ratio and blade thickness. These were three-bladed propellers, the blades being elliptical. Careful analysis of the results of these experiments shows that they may be expressed with close approximation by formulæ, which, though somewhat complicated, are not too inconvenient for use.

Let P denote the horsepower being absorbed by the propeller.

Let T denote the thrust of the propeller in pounds. (Kilograms in metric system.)

Let V denote speed of advance of the propeller (not the ship) through the water, in knots.

Let d denote diameter of propeller in feet. (Meters in metric system.)

Let p denote pitch in feet of the driving face. (Meters in metric system.)

Let a denote pitch ratio or $p \div d$.

Let w denote mean-width ratio or (mean width of blade $\div d$).

Let t denote blade-thickness fraction, or (thickness of radial blade section at axis if lines of face and back are extended) $\div$ diameter.

Let R denote revolutions per minute.

Then

$$\text{Slip, } s, = \frac{pR - 101\tfrac{1}{3}\,V}{pR}.$$

In metric units,

* From a paper presented at the International Engineering Congress, San Francisco, Cal., September, 1915. (For previous installment see October, 1915, issue.)

† Chief Constructor, United States Navy

‡ "The Speed and Power of Ships," by D. W. Taylor, 1910.

* "Speed and Power of Ships." By D. W. Taylor, 1910.

$$s = \frac{pR - 30.9\,V}{pR}.$$

For blade widths and thicknesses and working slips found in practice, we have for elliptical three-bladed propellers, with blades of ogival section, advancing through undisturbed water

$$T = \frac{d^2v^2}{(1-s)^2} \times \frac{s + \frac{.057}{w} - .0785 + a\left(.015 - \frac{.0175}{w}\right) + \left\{2.54 - 32\,(w-.15)^2\right\} t}{.2 + (a-.5)\,\frac{9.8 - 16\,w}{9}}$$

In metric units,

$$T = 4.883\,\frac{d^2v^2}{(1-s)^2} \times \frac{s + \frac{.057}{w} - .0785 + a\left(.015 - \frac{.0175}{w}\right) + \left\{2.54 - 32\,(w-.15)^2\right\} t}{.2 + (a-.5)\,\frac{9.8 - 16\,w}{9}}$$

For working slips, say, between $s = 0.2$ and $s = 0.35$ and for values of w between, say, 0.25 and 0.40, this formula represents almost exactly the results of the model-

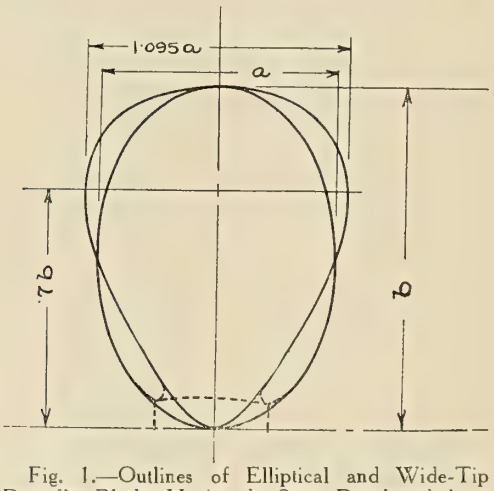


Fig. 1.—Outlines of Elliptical and Wide-Tip Propeller Blades Having the Same Developed Area and Mean Width Ratio

propeller experiments at the U. S. Model Basin. While not quite so close, the agreement with a series of experiments made by Mr. R. E. Froude* with elliptical propellers is quite close enough for practical purposes. While the formula includes the mean-width ratio, w, the influence of this factor is comparatively small. Thus in Froude's experiments, above referred to, an increase of some 10 percent in mean width, other circumstances being unchanged, results in an increase of only about 1 percent in thrust.

While this semi-empirical formula applies strictly to elliptical blades, the shape may depart materially from the elliptical and the formula will still apply fairly well. Fig. 1 shows, contrasted with an elliptical blade, an extremely wide-tipped blade of the same area experimented with by Froude. The thrust from the wide-tipped blade is only some seven or eight percent greater than that given by the formula for the elliptical blade.

When we undertake to devise a semi-empirical formula for the power absorbed by an elliptical-bladed propeller, it is found that variation of blade width affects it so little that, for practical purposes, we need not consider w in the formula for power, which is

$$P = \frac{0.88 + \frac{0.69}{a} + \left(1.3 + \frac{10.3}{a}\right) t}{1-s} \left(\frac{pR}{1{,}000}\right)^3 \frac{d^3}{p}$$

We may write this

$$\frac{P\,(1-s)}{\left(\frac{R}{100}\right)^3} = \left\{0.88 + \frac{0.69}{a} + \left(1.3 + \frac{10.3}{a}\right) t\right\} \frac{d^3p^2}{1{,}000} = C, \text{ say.}$$

In metric units,

$$\frac{P\,(1-s)}{\left(\frac{R}{100}\right)^3} = \left\{0.88 + \frac{0.69}{a} + \left(1.3 + \frac{10.3}{a}\right) t\right\} 0.380\,d^3p^2 = C$$

Where C is constant and known for a given propeller.

Derived from this, the equation,

$$1 - s = \frac{C}{P}\left(\frac{R}{100}\right)^3$$

will be found very useful for approximating the real slip, as contrasted with the apparent slip, when analyzing trial results.

Examination and consideration of the above formulæ and of the experimental results which they represent show, as to be expected, that for cases as found in practice the slip is the main factor and the variation of slip incident to a permissible variation of revolutions for a given power will overshadow the effect of usual variation of blade width and thickness.

For a final approximation, then, for design purposes, the experimental results give the following for three-bladed propellers with elliptical blades, mean-width ratio from 0.25 to 0.40 and blade-thickness fractions such as are found in practice.

For best efficiency

$$d = \frac{(PV)^{1/5}}{R^{2/3}}\left(63.5 - 50\,\frac{V^{5/2}}{RP^{1/2}}\right)$$

In metric units,

$$d = \frac{(PV)^{1/5}}{R^{2/3}}\left(19.1 - 15.3\,\frac{V^{5/2}}{RP^{1/2}}\right)$$

Pitch ratio,

$$a = 0.74 + 3.3\,\frac{V^{5/2}}{RP^{1/2}}$$

Efficiency

$$= 0.768 - 0.01\,\frac{RP^{1/2}}{V^{5/2}}$$

For propellers with smaller diameter and working with .025 less efficiency

$$d = \frac{(PV)^{1/5}}{R^{2/3}}\left(57 - \frac{50\,V^{5/2}}{RP^{1/2}}\right)$$

In metric units,

$$d = \frac{(PV)^{1/5}}{R^{2/3}}\left(17.4 - \frac{15.3\,V^{5/2}}{RP^{1/2}}\right)$$

Pitch ratio,

* "Results of Further Model Screw Propeller Experiments," Transactions of the Institution of Naval Architects, 1908.

$$a = 1.08 \frac{3.3\, V^{5/2}}{RP^{1/2}}$$

Efficiency

$$= .743 - .01 \frac{RP^{1/2}}{V^{5/2}}$$

For propellers with larger diameter and working with .025 less efficiency

$$d = \frac{(PV)^{1/6}}{R^{2/3}} \left(70 - \frac{50\, V^{5/2}}{RP^{1/2}} \right)$$

In metric units,

$$d = \frac{(PV)^{1/6}}{R^{2/3}} \left(21.3 - \frac{15.3\, V^{5/2}}{RP^{1/2}} \right)$$

Pitch ratio,

$$a = .52 + \frac{3.3\, V^{5/2}}{RP^{1/2}}$$

Efficiency

$$= .743 - .01 \frac{RP^{1/2}}{V^{5/2}}$$

For four-bladed propellers, with close approximation the diameter, pitch and efficiency are, respectively, 0.94, 0.98 and 0.96 of those for three-bladed propellers; while for two-bladed propellers they are 1.035, 1.01 and 1.02.

The preceding formulæ assume that the law of comparison applies fully and that the speed of advance, V, of the propeller through the water is known. If the law of comparison did apply to all cases, with the data which have now become accumulated from model-propeller experiments, the propeller problem would, in practically all cases, be capable of a solution in advance sufficiently close for practical purposes, if not of scientific accuracy. As a matter of fact, the law of comparison does apply with close approximation to the majority of practical cases. For the propeller, however, as for the ship, in order that the law may apply exactly, the pressures should be in the same ratio as the dimensions. This, of course, is not the case. The model propeller is tested under a head of, perhaps, 1 foot (.3 m.), due to submergence, and 34 feet (10.4 m.), equivalent to the atmospheric pressure. The full-sized propeller operates with submergence in proportion to its size, but the head due to atmospheric pressure is the same for the full-sized propeller as for the model. Hence we find that with some full-sized propellers, particularly the high-speed type, the now well-known phenomenon called "cavitation" occurs, while there is no indication of it with the models of such propellers. Since Barnaby first described cavitation,* as observed in the trials of the *Daring* many years ago, there has been much study and thought given to it, but the only process, in practice, which has been generally satisfactory in reducing or avoiding it is increase in blade area and reduction of the thickness of the blades (particularly at the leading edge) to the limit permissible by practical considerations.

The many endeavors to devise a form of blade section of moderate width which the water will continue to hug when the speed of the blade through the water is extreme, have not heretofore been strikingly successful. It is possible that they may show some success in the future, enabling the propeller designer to adopt a type of blade somewhat more efficient than the very wide blade, whose friction is apt to be excessive. Even so, the gain would not be very great, as the loss of efficiency due to the broad blade is not very serious.

Attempts accurately to define the region where cavitation is to be expected have not, hitherto, been very successful. The limiting thrust per unit of projected area originally proposed by Barnaby has been materially exceeded in some cases without cavitation, while in other cases cavitation has been found with much lower thrusts. The same statements virtually apply to tip speed, which is, perhaps, a somewhat better criterion. Here again, the shape of the blade section is a matter of primary importance. For a given type of blade section we may adopt either tip speed or thrust as a criterion, with fairly satisfactory results, provided we do not go beyond the limits which have been found satisfactory for the same type of blade in other cases.

Attempts have been made to investigate cavitation with models. Sir C. A. Parsons has made experiments along this line with hot water, or in a partial vacuum. The idea, in each case, was to reduce the pressure around the model in proportion to its size. Little or nothing of value in defining the field of, or in avoiding, cavitation, has been published as a result of these experiments.

The formulæ given, derived from model experiments, assume, as already stated, that V, the speed of advance through the water, is known. When we come to apply the propeller at the stern of the ship we do not know, as a rule, the speed of advance through the water. The propeller acts in the wake of the ship and in most cases the wake is positive, which means that the speed of advance of the propeller through the disturbed water in which it works is less than the speed of the ship. Model experiment investigations have been made by R. E. Froude,* and notably by W. J. Luke,† of late years, using a model propeller behind the model of the ship. This method enables not only the wake but the thrust deduction, or suction of the propeller upon the stern of the ship, to be investigated. From Luke's results we may roughly express the wake as follows:

Let k denote the actual wake fraction, the virtual uniform following current in which the propeller acts being equal to kV, where V is the speed of the ship. Let b denote block coefficient. Then for single screws $k = 0.05 + 0.5b$. For twin screws $k = 0.2 + 0.55$b. These are very rough approximations, as reference to Luke's papers will show that the wake varies with the position of the screws and also with their direction of rotation. The wake means a gain in efficiency, whereas the thrust deduction means, of course, a loss as compared with the case where the propeller is driving what R. E. Froude has happily called a "phantom ship," namely, a ship which offers the resistance of the actual ship but does not disturb the water. It is usually assumed that the wake gain and thrust deduction virtually offset each other as regards efficiency. This is, actually, not exactly the case, and this whole matter of wake factor and thrust deduction is one needing a thorough clearing up. The assumption that the wake is uniform is, of course, simply a working hypothesis for convenience, the actual wake being far from uniform. This was demonstrated many years ago in some experiments made by Mr. George A. Calvert.‡ Experiments recently made at the U. S. Model Basin, with special apparatus, throw additional light upon this matter.

When fluid is flowing past a sphere, or a sphere is advancing into still water, the excess pressure on the sphere (above that due to its submergence) immediately around

* The Formation of Cavities in Water by Screw Propellers at High Speed," Sydney W. Barnaby, Transactions of the Institution of Naval Architects, 1897.

* "A Description of the Method of Investigation of Screw Propeller Efficiency," Transactions of the Institution of Naval Architects, 1883. "The Determination of the Most Suitable Dimensions for Screw Propellers," Transactions of the Institution of Naval Architects, 1886.

† "Experimental Investigations of Wake and Thrust Deduction Values," Transactions of the Institution of Naval Architects, 1910. "Further Experiments on Wake and Thrust Deduction," Transactions of the Institution of Naval Architects, 1914.

‡ "On the Measurement of Wake Currents," Transactions of the Institution of Naval Architects, 1893.

the axis of advance is equivalent to the velocity head of advance, and the pressure at 90 degrees from the axis of advance is below the natural pressure due to submergence. Around some circle between the two, the pressure in the moving water around the sphere will be the same as the pressure in the undisturbed water at the same depth below

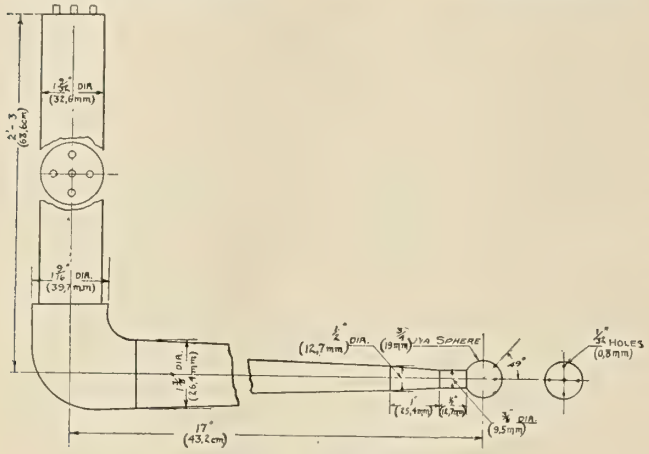


Fig. 2.—Sketch of Wake Meter Tube

the surface. In theory, this circle is 41° 49′ away from the axis. As a matter of fact, careful experiments with a ¾-inch (19 mm.) sphere showed that, due to friction or some other cause, the actual distance from the axis was almost exactly 49°. A ¾-inch (19 mm.) sphere, Fig. 2, was made with five holes 1/32 inch (0.8 mm.) in diameter, one in the axis, two on a horizontal line 49° away from the axis, and two on a vertical line 49° away from the axis. Each hole was connected with a vertical glass tube showing head and change of head. The sphere was on the end of a horizontal arm, which was carried, in turn, by a vertical tube sufficiently far away not to affect the motion around the sphere. The apparatus was calibrated from the Model Basin carriage, by advancing it with the axis of the tube or the central hole at various angles from both the vertical and horizontal. It was found that the difference between pressure head measured at the center hole and the average head of the other holes indicated the speed with great accuracy, and the differences of head shown in the tubes connected with the other holes enabled the angle of advance to be determined accurately, the error being no greater than a fraction of a degree. It was also found that this sphere could be placed so that its center would be within 1½ inches or so of a plane without the motion around it being materially affected, and, when placed closer, the variation from normal conditions could be calibrated. The sphere then being placed in any position near the stern of a model, whose speed through the water is known, and readings taken of the heights of water in the tubes connected with the various holes, we can determine the actual speed and direction of flow of the water.

Fig. 3 shows for a model the directions with reference to the axis of the vessel under way and the amounts of the wake currents at the points indicated. Since friction is relatively greater in a model than in a full-sized vessel, the amounts and directions of the wake of a full-sized vessel would differ slightly from those around the model, but it is not likely that the difference would be material.

Section B would be a fairly usual location for the propeller of this vessel, the circle indicating a propeller of 8 feet diameter for the full-sized vessel. It is seen, then, that the blades of such a propeller encounter most variable conditions during a revolution, the axial or fore and aft velocity of the wake varying from 4 to 23 percent. These axial velocities are for a vessel which has fine lines aft;

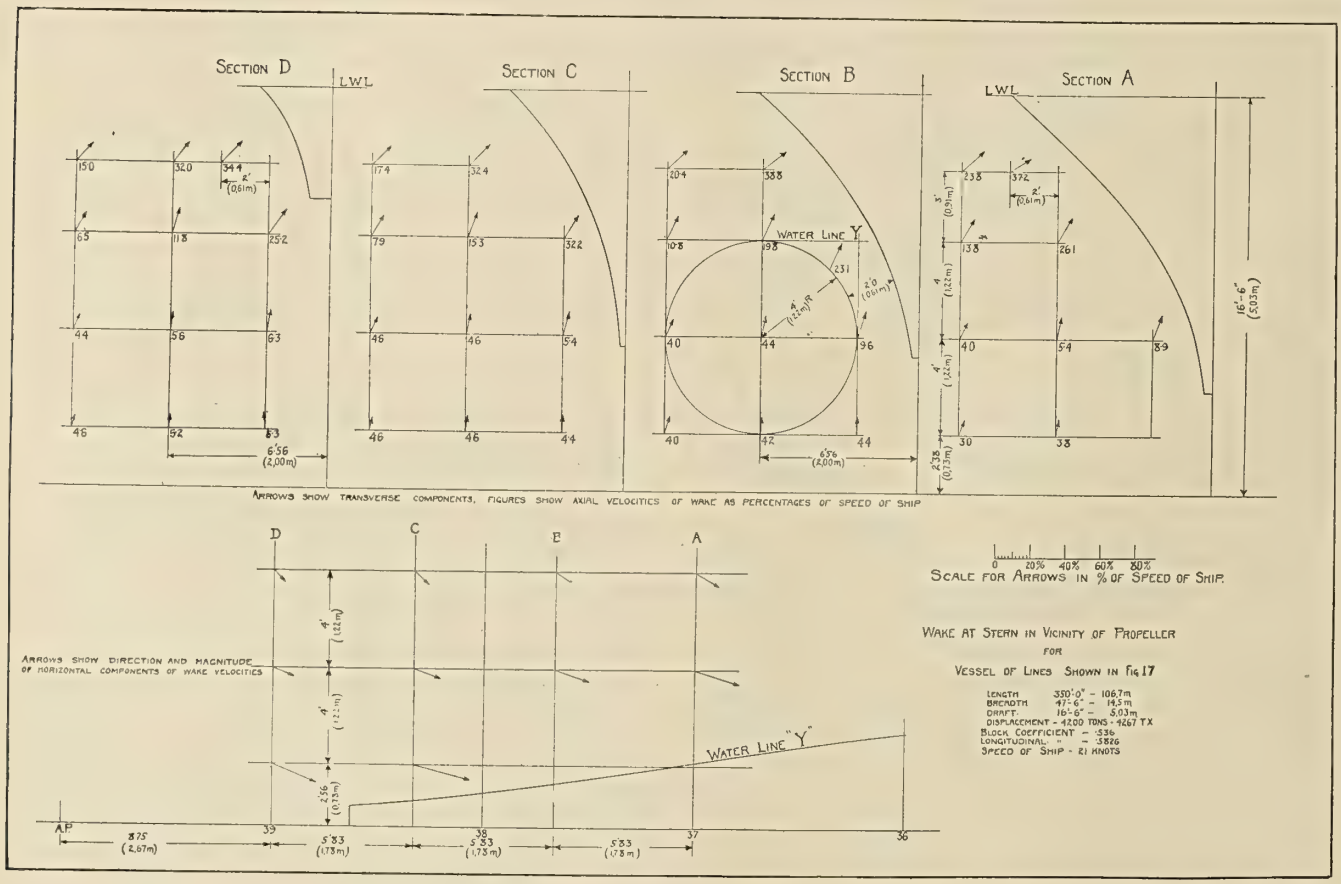

SECTION D
LWL
SECTION C
SECTION B
WATER LINE Y
SECTION A
LWL
ARROWS SHOW TRANSVERSE COMPONENTS, FIGURES SHOW AXIAL VELOCITIES OF WAKE AS PERCENTAGES OF SPEED OF SHIP
SCALE FOR ARROWS IN % OF SPEED OF SHIP
ARROWS SHOW DIRECTION AND MAGNITUDE OF HORIZONTAL COMPONENTS OF WAKE VELOCITIES
WATER LINE "Y"
WAKE AT STERN IN VICINITY OF PROPELLER FOR VESSEL OF LINES SHOWN IN FIG 17
LENGTH 350'-0" - 106.7m
BREADTH 47'-6" - 14.5m
DRAFT 16'-6" - 5.03m
DISPLACEMENT - 4200 TONS - 4267 TX
BLOCK COEFFICIENT - .536
LONGITUDINAL " - .3826
SPEED OF SHIP - 21 KNOTS

Fig. 3

for a full vessel, the wakes would be greater and the variations greater. It would seem evident that when the propeller must work under the conditions indicated by Fig. 3, fine-spun theories as to angles of slip, etc., in still water, are quite irrelevant as regards actual conditions. Incidentally, it may be remarked that the wakes indicated at distances from the hull, where the frictional wake would have little or no effect, agree very well in a general way with what would be expected from theoretical considerations of stream-line flow.

Some recent experience with United States vessels has emphasized the desirability of having strut arms parallel to the lines of natural flow of the water, and the apparatus referred to enables the proper angles to be readily determined. Sometimes these are far from what we would make them, if evolved from our inner consciousness.

[In a later issue the third and final instalment of this paper will be published, dealing with the subject of rolling of ships. In recent years there has been renewed interest in this important subject, and the investigations made of it, instead of being of academic interest only, are of great practical interest and value.—Editor.]

Modern Submarines in War and Peace—VIII

Application of the Submarine to Commercial Enterprises—Cargo-Carrying Submarines—Sea Bottom Exploration and Wreck Salvage

BY SIMON LAKE

The quite unexpected success of the naval submarine will perhaps now give financiers greater confidence in the value of the submarine for a great variety of commercial purposes, where its field for exploration, constructive work, and recovery of lost ships and cargoes, submarine mining and recovery of foodstuffs and other valuable natural products of the sea will make it, when it reaches its full development, one of the most important inventions of the age. All it needs is capital and experienced engineers and operators to take from the sea what the misfortunes of man (in the way of lost ships and cargoes) and nature has provided in such abundance.

Cargo-Carrying Submarines

The author has frequently been asked as to what is the limit in size of the submarine. That is a difficult question to answer, as it depends so much upon the depth of submergence, speed desired, etc. The usual circular-formed vessel, if designed to meet the present United States Government requirements of 200-feet depth of submergence,

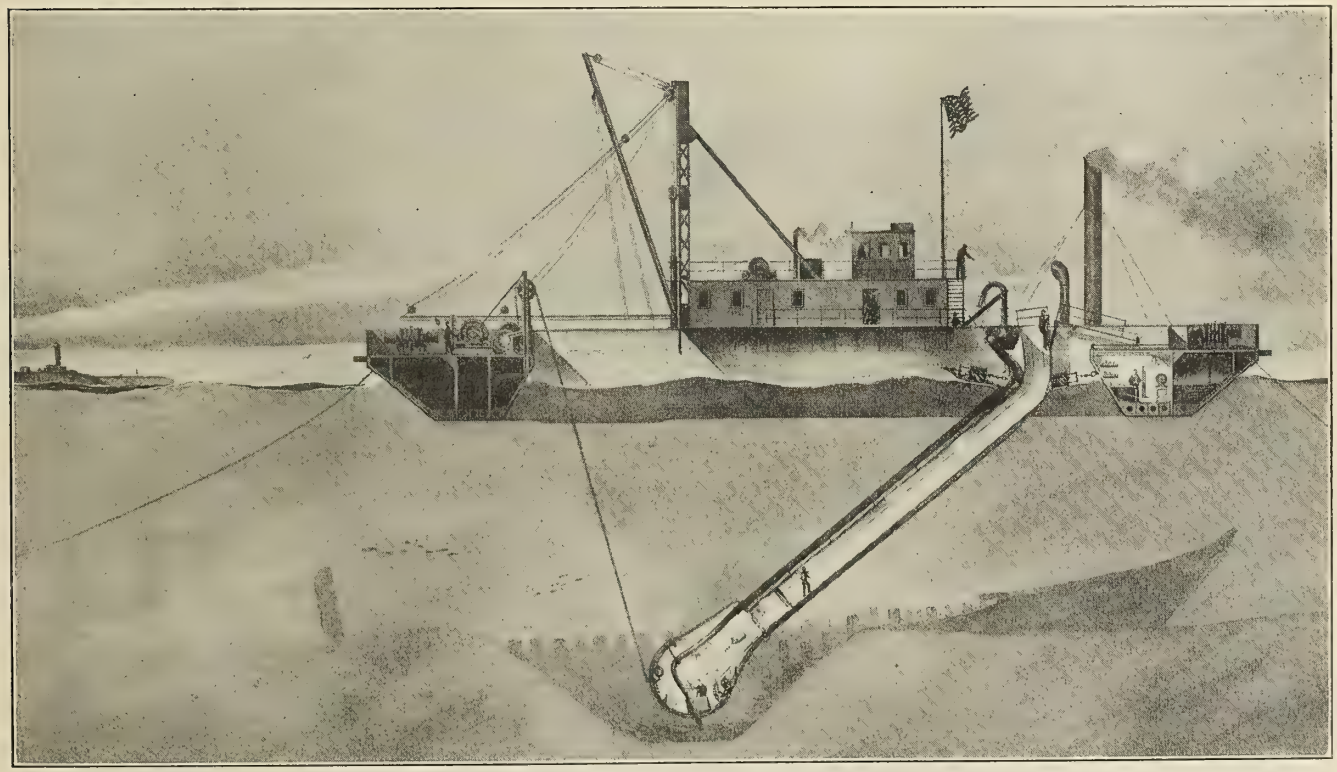


Fig. 55.—Semi-Submergible Wrecking Apparatus

The submergible tube has the diver's operating compartment shut off from the rest of the apparatus by means of an air lock which permits passage from the surface vessel and normal air pressure to the diver's compartment, where the air is under pressure equal to the compartment's depth of submergence, when the diver's exit door is open. The above illustration shows clearing out a vessel, from sand, by the use of a powerful sand pump to enable the salvors to get at the cargo. The same method would be employed in pumping up a cargo of coal. With this apparatus it is only necessary to "walk down stairs," so to speak, to go down into a sunken ship, and if necessary to send divers out they are convenient to tools and are not subject to the dangers incident to entanglements of life line and air hose when going down into the murky darkness from a surface vessel by jumping into the seas. No matter how much the surface vessel may jump about on the sea, the compartment at the bottom is still.

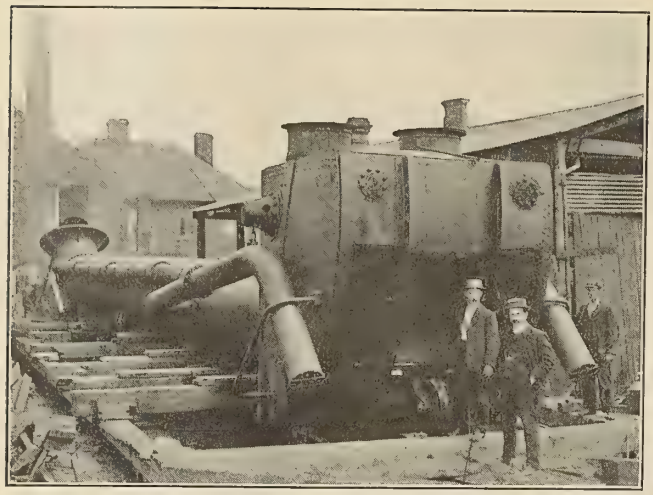


Fig. 56.—Experimental Submarine Wrecking Tube, Built by Mr. Lake in England to prove out the practicability of using an apparatus of this kind in connection with a surface vessel in wrecking operations

would need to be built of very heavy plates and frames, and would probably be of too great draft to enter our harbors if built to carry as much as 5,000 tons cargo. A submarine vessel built up of nests of cargo-carrying, light-weight circular tanks of comparatively small diameter, surrounded by a ship-shaped form of hull, has been patented by the author, which lends itself to much larger construction, and a greater safe depth of submergence on less weight of material and draft of ship.

The multiplicity of units would permit of a number of them being punctured by shell fire while in surface cruising condition without sinking the ship. If the present European war continues for a considerable period of time we shall probably see some form of cargo-carrying submarine constructed to be used as blockade runners.

Sea-Bottom Exploration

Investigation of the geological formation of sea bottoms, the flora and fauna of the sea, will be greatly assisted by bottom-creeping submarines. Fitted with powerful searchlights and moving-picture cameras, actual sea-bottom conditions may be reproduced up to depths of 1,000 feet or more. The author, in 1898, succeeded in taking photographs of fish through the windows of the *Argonaut* by means of an ordinary kodak, and last year the Williamson brothers showed in moving-picture houses throughout the country some wonderful submarine moving pictures they had secured by the use of their collapsible submarine tube.

In Southern waters it is possible to see considerable distances, and in certain localities the great variety of brilliantly colored fish swimming about among the varicolored kelp, sea grasses and corals. On a sunshiny day, when the play of lights from the undulating underneath surface of the water, which resembles a rippling, polished mirror, is a "sight" never to be forgotten and is well worth a thousand-mile voyage to witness. The writer is now engaged on the plans of a passenger-carrying submarine that will carry fifty passengers, and is to be especially built to enable people to view these "wonders of the

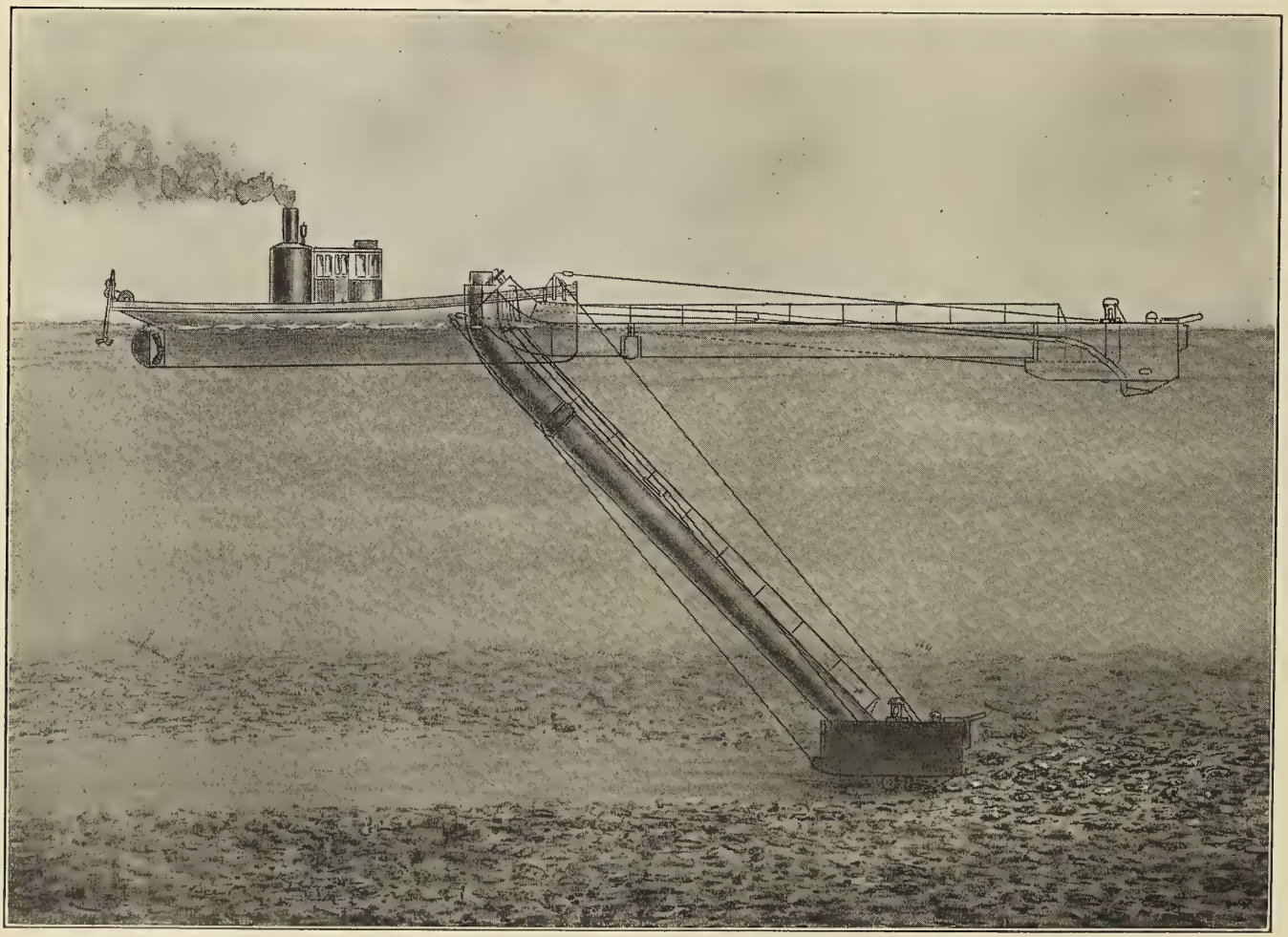


Fig. 57.—Submarine Boat Attached to Surface Vessel with Communicating Access Tube

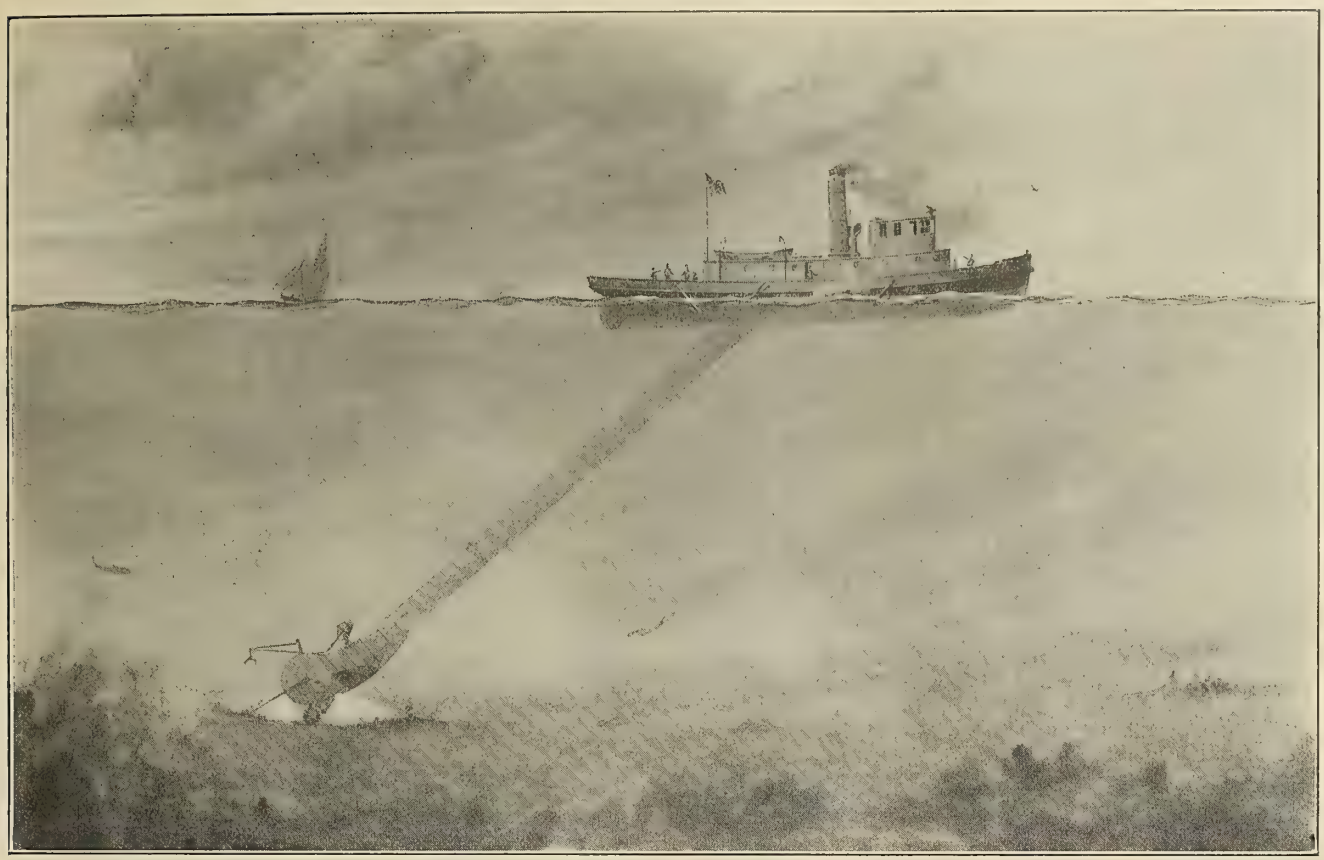


Fig. 58.—Submarine Apparatus for Use in the Pearl and Sponge Fisheries

The submarine tube has a working compartment at one end and is provided with water ballast tanks so that it may be submerged to the bottom. It is weighted so as to give sufficient traction to pull the surface vessel if desired, or the surface vessel may pull it over the bottom. A traction wheel to propel the compartment may be turned in any direction and is driven by a motor inside the operating chamber. Air pressure may be admitted to the operating chamber to equalize the hydrostatic pressure outside of the compartment, and then a door may be opened in the bottom and no water will enter the compartment. In this manner articles may be gathered directly off the bottom by hand or divers may leave the compartment by donning the regulation dress, or in deep water grabs or rakes may be manipulated from the interior and the product deposited in carriers which take the material to the surface, without the necessity of the operator undergoing pressure.

deep" which have heretofore been only partially available to native divers of the Southern seas or the few who have penetrated the depths for short intervals of time in the diving dress.

In this "sight-seeing" passenger-carrying "automobile" submarine it will be possible for the passengers to sit comfortably at their respective windows and view the coral formations and the submarine forests of kelp and other monster sea grasses, some of which grow to over one hundred feet in length, and watch the play of the great variety of fishes which abound among them.

Those of my readers who have viewed the submarine gardens at Bermuda, Nassau, or in the Caribbean Sea, or Catalina Islands, or San Diego, on the Pacific, through glass-bottom boats may have some comprehension of the beautiful sights that await those who may be privileged to actually go down among the "denizens of the deep" and see the full beauties of the most beautiful of nature's gardens in their natural perspective and coloring, instead of only the "plan" view, so to speak, when viewed from above.

Submarine Wreck-Finding and Recovery Boats

In searching for sunken vessels, two boats are used, of the same general type as the "hydrographic submarine." When a wreck is located, divers go out and examine it. If it is concluded that she has cargo on board worth salving, her location is plotted on the chart, and then the recovery boats are sent out to remove the cargo. The writer has done much experimental work in locating sunken wrecks and recovering their cargoes. In 1898, 1899 and 1900 the *Argonaut* and special wreck-finding apparatus were used in this experimental work. Numerous wrecks were found and a number of cargoes were profitably recovered, notwithstanding the fact that the apparatus used was crude and experimental. In 1901 I was called from this line of work to take up the construction of submarine torpedo boats, and have been too busy ever since building for the United States and foreign governments, and have not had the time and opportunity to push on this very interesting phase of submarine work.

Searching for sunken vessels is, perhaps, the most interesting of all submarine work. It is like fishing. One is always on the "qui vive" for a "bite." There is hardly a location along our coast or in Long Island Sound that does not have a tradition about lost treasure ships, and every time one gets a "bite"—that is, our lines get fast to some sunken object, excitement runs high in the expectation of some valuable find. In my experimental work in the vicinity of Bridgeport, Conn., we located sixteen sunken vessels, the great majority of them containing coal, which we recovered at a cost of about fifty cents (2/1) per ton. Most of these vessels had been sunk a long time. Only a few of them were known by name, and some had evidently been sunk many years. One that we searched for, for several months, had a cargo of copper ore and copper matte which was quite valuable. We finally found her several miles away from where people testified they saw her disappear.

Somewhere off Bridgeport lies the wreck of the old Sound steamer *Lexington*. Legend has it she has a for-

tune in her safe. Many a ship has been sunken in the waters about Hell Gate. Search was carried on there for years for the old British frigate *Hussar,* which struck on Pot Rock and sank during the Revolutionary War. Tradition has it that she had four million dollars (£820,000) in gold on board to pay off the British troops, and that she carried this treasure to the bottom with her. There is a cargo of block tin somewhere in a sunken barge off the Battery, and many a ship with valuable cargoes lies along the coast from New Foundland to Key West. The time will come when many of these ships will be found and such of their cargo as is still valuable will be salved.

Salving a sunken cargo is not a difficult engineering feat, providing the proper apparatus is at hand. It is the novelty of the enterprise and the mystery surrounding submarine work that makes it seem so difficult to the layman. Diving as heretofore conducted has been difficult and dangerous work, and only the strong could stand the hardships connected with it. The advent of proper machinery and scientific methods, however, will clear away many of the hardships and dangers connected with salving a sunken cargo, and more experience and proper apparatus will prove that certain cargoes may be removed from sunken ships in moderate depths with almost as much rapidity as they can be lifted from the hold of a vessel alongside of a dock. Take anthracite coal, for instance. With a six-inch pump, on the old *Argonaut,* I have transferred fifteen tons of nut coal from a sunken barge to a sunken freight-carrying submarine in nine minutes. A turn of the air valve then sent the sunken freight boat to the surface. The coal was transferred while all the boats were submerged in seven fathoms of water. It was this kind of experimental work which has enabled me to devise apparatus which will undoubtedly operate successfully on a much larger scale, as explained below.

The wrecking barge which is illustrated on page 75 is to be fitted with a 15-inch coal pump, which should recover any of the usual household grades of anthracite coal at the rate of ten tons per minute. All the manipulation of the suction pipe is done by power, instead of by hand by divers, as heretofore. Especially designed compensating hoists, operated from a pressure-resisting compartment right down on the job, permits breaking out cargo in comparatively rough weather. The divers operate within a few feet of a compartment and air lock, whereby they largely avoid the dangers incident to entanglement of life line and air hose, and whereby they can become gradually prepared for deep-water work by undergoing "stage" compression and decompression, which is the proper method, according to the latest scientific research. All the time which divers formerly spent being swung back and forth by the currents and jerked up and down by the action of the sea is avoided and the compression and decompression periods are undergone while at rest in a well-lighted and comfortable compartment. Diving work may therefore be carried on at much greater depths and for much longer periods of time than heretofore.

Statistics have been published to show that practically the entire commerce of the world sinks in every twenty-five years. A report from Berlin states that "734 enemy ships, with a tonnage of nearly one and half millions, have been sunk by the Germans, of which 568 of these ships were sunk by submarines."

Most of these vessels were sunk in the North Sea or the English Channel, where the water is comparatively shallow, and where many of the cargoes can undoubtedly be recovered with the proper apparatus. The loss of ships is such a common occurrence that little attention is paid to them except when their loss is accompanied by great loss of life, as was the case with the *Titanic,* the *Monroe,* the *Empress of Ireland,* or the *Lusitania.*

(To be continued.)

Pitches for Screw Stays

The following table gives the pitches for screwed stays for supporting flat plates, as worked out according to the United States Government rule in which the following formula is used:

$$\text{Pitch} = \sqrt{\frac{112 \times T^2}{P}},$$

where T = thickness of plate in 16th of an inch,
P = working pressure in pounds per square inch,
112 = constant for plates 7/16 inch and under.

The same formula is used for plates 15/32 inch and over by substituting 120 as the constant in place of 112 used in the above formula.

The areas and pitches in this table will be found to be correct, or, in some cases the pitches may be a little small, which favors the work.—*The Boiler Maker.*

Table of Pitches for Screwed Stays Supporting Flat Plates (U. S. Government Rules)

Plate	1/4		9/32		5/16		11/32		3/8		13/32		7/16		15/32		1/2		17/32		9/16		5/8		11/16	
W.P.	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch	Area	Pitch
50 #	35.84	5.98	45.36	6.73	56.00	7.48																				
60	29.86	5.46	37.80	6.14	46.66	6.83																				
70	25.60	5.06	32.40	5.69	40.00	6.32	48.40	6.95	57.60	7.58	67.60	8.22	76.98	8.77												
80	22.40	4.74	28.35	5.32	35.00	5.91	42.35	6.50	50.40	7.10	59.15	7.68	67.35	8.20												
90	19.92	4.46	25.20	5.02	31.11	5.57	37.64	6.13	44.80	6.69	52.58	7.25	59.86	7.73												
100	17.92	4.23	22.68	4.76	28.00	5.29	33.88	5.82	40.32	6.35	47.32	6.87	53.88	7.34	67.50	8.21	76.80	8.76	86.70	9.31	97.20	9.86	120.	10.95	145.	12.04
110	16.29	4.03	20.62	4.54	25.46	5.04	30.80	5.55	36.66	6.05	43.00	6.55	48.98	6.99	61.87	7.83	69.82	8.35	78.82	8.87	88.36	9.40	109.	10.44	132.	11.49
115	15.59	3.95	19.72	4.44	24.35	4.93	29.46	5.42	35.07	5.92	41.15	6.41	46.85	6.84	58.69	7.66	66.79	8.17	75.40	8.68	84.53	9.19	105.	10.24	126.	11.23
120	14.94	3.86	18.90	4.34	23.34	4.83	28.23	5.31	33.60	5.79	39.44	6.28	44.83	6.69	56.25	7.50	64.00	8.00	72.25	8.50	81.00	9.00	100.	10.00	122.	11.00
125	14.33	3.78	18.14	4.26	22.40	4.73	27.10	5.20	32.26	5.68	37.86	6.15	43.11	6.56	54.00	7.34	61.44	7.83	69.36	8.32	77.75	8.81	96.00	9.79	116.	10.77
130					21.54	4.64	26.06	5.10	31.02	5.57	36.40	6.03	41.45	6.43	51.92	7.20	59.08	7.68	66.70	8.16	74.77	8.64	92.00	9.59	112.	10.58
140					20.00	4.47	24.20	4.92	28.80	5.36	33.80	5.81	38.49	6.20	48.22	6.94	54.86	7.40	61.93	7.87	69.43	8.33	86.00	9.27	104.	10.19
150					18.66	4.32	22.58	4.75	26.88	5.18	31.54	5.61	35.92	5.99	45.00	6.70	51.21	7.15	57.80	7.60	64.80	8.05	80.00	8.94	97.00	9.85
160									25.20	5.02	29.56	5.43	33.68	5.80	42.19	6.49	48.00	6.93	54.19	7.36	60.75	7.79	75.00	8.66	90.00	9.52
170									23.72	4.87	27.83	5.27	31.70	5.63	39.71	6.30	45.18	6.72	51.00	7.14	57.18	7.56	71.00	8.42	85.00	9.22
180													29.94	5.47	37.50	6.12	42.66	6.53	48.17	6.94	54.00	7.35	67.00	8.17	81.00	9.00
190													28.36	5.32	35.53	5.96	40.43	6.36	45.64	6.75	51.16	7.15	63.00	7.93	76.00	8.71
200													26.94	5.19	33.75	5.81	38.40	6.19	43.35	6.58	48.60	6.97	60.00	7.74	72.00	8.47

Letters from Marine Engineers

Discussion of the Design and Handling of Marine Engines, Boilers and Auxiliaries—Breakdowns at Sea and Repairs

This department is open to all readers of the magazine for the discussion of affairs in the engine room. All letters published are paid for at regular rates. Your ideas or experiences will be mutually helpful and interesting to other engineers. Write your letter now.

Coaling a Liner at Nagasaki

Although the coaling of the large transatlantic liners in New York harbor is carried out by machinery and over 8,000 tons of coal have been placed in a ship's bunkers in nineteen hours, nevertheless the coaling of a liner at Nagasaki, Japan, is quite a different matter, as the work

Bunkering the *Shinyo Maru* at Nagasaki—3,500 Tons of Coal Taken on Board in Twenty-four Hours

is done entirely by hand, as shown in the accompanying photograph. This photograph was taken from the deck of the Japanese liner *Shinyo Maru,* of 13,384 gross tons, while coal was being taken on board at the rate of 3,500 tons in 24 hours, and gives an excellent idea of the great number of men required for transferring the coal from the small barges alongside to the ship's bunkers.

Manila, P. I. C. N. CONOVER,
Chief engineer, U. S. Coast and Geodetic Survey.

Checking Up the Coal Consumption

While every item of expense in the operation of a line of steamers is no doubt closely scrutinized by the general manager, perhaps that of coal receives the closest attention of any, because it is the biggest single item of expense incurred. I found on taking charge of the engine room of a bay steamer that a daily report of the coal consumed was required to be turned in to the office at the end of each round trip. This was something of a problem, for we had no accurate means for determining just what that was. Of course we knew about what the average per day was by the amount of coal delivered to us. This was calculated at the end of each month, the total consumption being divided by the number of days in the month. But this did not tell us what we did from day to day with varying conditions, and naturally it was impracticable to go into the bunker with a tape line and measure it up every morning. Sometimes if the coal bills were "up" we even felt that perhaps the weights were short. To have the coal weighed in the cars at the yard costs one dollar (4/2) a car, and, unless so weighed it was sold to us by the mine weight, and, as the mines were more than three hundred miles away, some of us received these figures with a mental reservation. The coal was wheeled aboard in barrows, and the barrows always counted and occasionally weighed, but this did not satisfy us entirely, and, besides, we wanted to get at the consumption from day to day, as stated above, and how it changed with varying conditions, the run being the same distance every night—about two hundred miles.

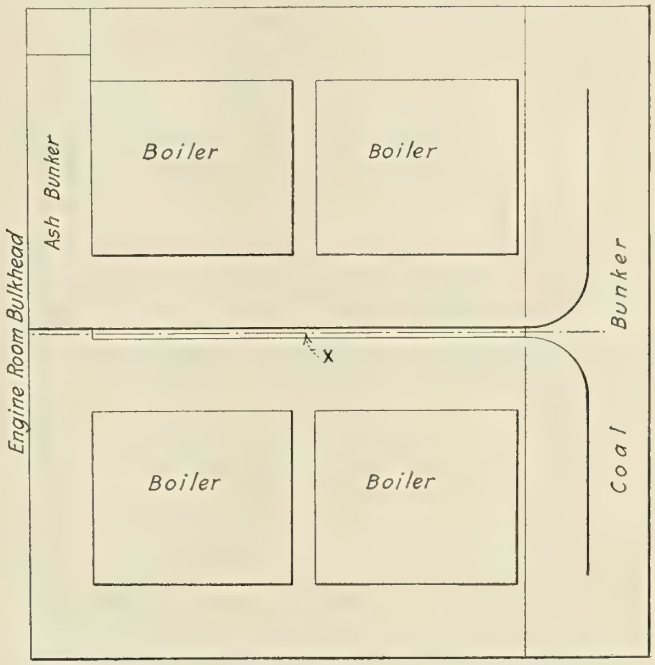


Fig. 1.—Diagram of Boiler Room, Showing Location of Trolley and Tripping Device

It had been customary for years to count the buckets of coal as brought out of the bunker, and this was what had been recorded on the daily report. These reports were seldom anywhere near the coal bill at the end of the month. For one reason it depended on a coal passer marking up

each bucket as it was brought out of the bunker. Of course, sometimes he marked it up, and sometimes he forgot it, and, for another reason, the location of the bunker door was such that the forward boilers could be fired directly with the coal that ran out on the floor when the bunker was filled. This latter was supposed to be accounted for by marking up a bucket for so many shovels. The result was that the count was so erratic as to be not worth recording. The chief engineer of our sister boat hit on a scheme for recording the count of buckets automatically with a register such as is used on street cars. A device was placed over the bunker door in such a way as to engage the trolley as it came out of the bunker and automatically pulled a wire that operated the register placed in the engine room. This arrangement proved unsatisfactory because the bunker had to be empty enough

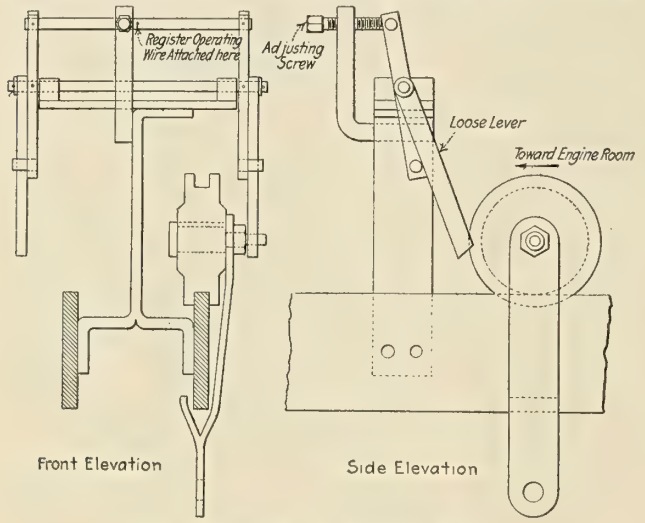


Fig. 2.—Sketch of Tripping Gear

to allow the bucket to enter the door or it would not work automatically, and had to be rung up by hand by the coal passer till considerable coal had been used, and of course it did not account for the coal on the fire-room floor that ran down to the forward boilers, as explained above, so that had to be counted as before.

While this arrangement was unsatisfactory, it rather tended to make the situation more acute, because it was presumed to be more accurate. After this had been tried and found wanting, it occurred to me that a much better scheme would be to install just such an arrangement as this, but to put the tripping gear over the track in the middle of the fire-room, and only to take account of the coal going to the after boilers. Assuming that all the boilers would consume about the same amount, the total used could be estimated to a very close approximation. As the coal could not very well be gotten to the after boilers except by the bucket, this arrangement was bound to work out pretty close in practice unless someone deliberately set about trying to fake the count, which is, of course, most unlikely.

The results were surprising, and also surprisingly accurate. The situation was immediately cleared up and there was no further argument as to whether we did or did not burn the coal. We are now accumulating some very interesting data. Our longest run without a stop is approximately 175 miles, and, running with the tide, we make around 56,000 revolutions at 85 or 86 revolutions per minute, and, running against the tide, we make around 65,000 to 66,000 revolutions at about 100 revolutions per minute. We have about eleven hours in which to make this run, and the difference in coal consumption between fair and adverse conditions not infrequently exceeds 75 percent. The sketches will enable anyone interested to install some similar arrangement, and, take it from me, a constant and reliable check on the coal consumption is a revelation to the uninitiated and, incidentally, it seems to be some satisfaction to the general manager.

Norfolk, Va. D. SAWYER,
Chief Engineer, steamship *Northland,* Norfolk and Washington Steamboat Company.

Indicator Test of Triple Expansion Marine Engine

As part of some thesis work at the Georgia School of Technology a test was run on the main engine of the steamship *City of St. Louis,* which is one of a fleet of coastwise steamers owned by the Ocean Steamship Company, plying between Savannah, Ga., and New York City.

The *City of St. Louis* is a combination freight and passenger steamer and a first class boat. She is 397 feet 9 inches in length, and measures 49 feet 6 inches at greatest width of beam. Two hundred and twenty passengers, including first and second class passage, can be comfortably accommodated, and her gross tonnage is 6,200. She was built by the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., during 1909-10, being launched in June of 1910.

She is equipped with four Scotch boilers, 14¾ feet in diameter, 11½ feet long, each having 280 stay tubes 3¼ inches in diameter, 7 feet long, and a heating surface of 8,500 square feet. The shells of the boilers are made of soft boiler plate steel 1¼ inches thick, having a tensile strength of 65,000 pounds per square inch. A double lap plate butt joint is used and the rivets are 1¼ inches in diameter, which gives 81 percent efficiency for the joint. These boilers have an allowable pressure of 180 pounds per square inch, but were tested when installed to 240 pounds per square inch, hydrostatic pressure.

A smokestack 8 feet in diameter and 60 feet long furnishes a natural draft for the boilers, which steamed well save at times, when the wind was from her stern.

Steam is conveyed from the boilers to the high-pressure cylinder through a 10-inch steel pipe, well lagged with a composition of asbestos and magnesia, and covered with polished thin sheets of steel.

In addition to these boilers, one donkey boiler is installed for use in case of emergency.

These boilers are supplied with water, from the hot well and feed water heater, through two pumps; also two injectors which are brought into play only when the steamship is in fresh water.

THE ENGINE

The main engine is of the vertical, triple expansion, condensing, marine type. The high-, intermediate- and low-pressure cylinders are 26 inches, 43 inches and 72 inches in diameter, respectively, with a common piston stroke of 4 feet. The cranks are set 120 degrees apart, operating in this sequence, high-pressure, intermediate-pressure and low-pressure. The steam from the low-pressure cylinder is discharged into a surface condenser through which sea water is circulated by a centrifugal pump.

The ratios of the volumes of the cylinders and receivers is as follows: high-pressure cylinder as unity, intermediate-pressure receiver 3, intermediate-pressure cylinder 2.75, low-pressure receiver 10.4, low-pressure cylinder 7.75.

Single-ported, balanced piston valves were used to control the admission of steam to the cylinders: one 12-inch valve for the high-pressure cylinder, two 12-inch valves for the intermediate-pressure cylinder and two 25-inch valves for the low-pressure cylinder, all having the same travel, viz., 7 inches, and controlled by the same type of valve mechanism, namely, the standard Stevenson link motion. The high-pressure valves take steam on the inside edge, while the intermediate-pressure and low-pressure valves take steam on the outside edge.

The crankshaft is 14 inches in diameter, supported by six babbit bearings 20 inches long, each being water cooled.

The piston is connected to the crosshead by a rod 6 inches in diameter. The crosshead reciprocates between two guides 30 inches apart and 20 inches wide, being 56 inches in length. Each is made of white babbit metal, water cooled and pitted over its surface to facilitate better lubrication.

The propeller is four-bladed, 16 feet in diameter, with a pitch of 19½ feet. The propeller shaft is 13⅞ inches in diameter, 106 feet long, made in four sections, supported every 20 feet by well-designed, water-cooled, babbit bearings. The shaft in each ran in a bath of oil, and showed no evidences of heating.

In addition to the main engine, there were in the engine room two 10-inch by 12-inch Westinghouse automatic engines, coupled directly to generators for developing electric current for lights and supplying the wireless apparatus. Among other apparatus in the engine room, besides the steam steering gear, was a common surface condenser, 5 feet in diameter and 11½ feet long, having about 4,000 square feet of condensing surface. The air pump connected to it was driven by a walking beam from the low-pressure crosshead. A centrifugal pump, two direct-acting steam pumps, a fire and sanitary pump, comprised the rest of the engine room equipment.

The Test

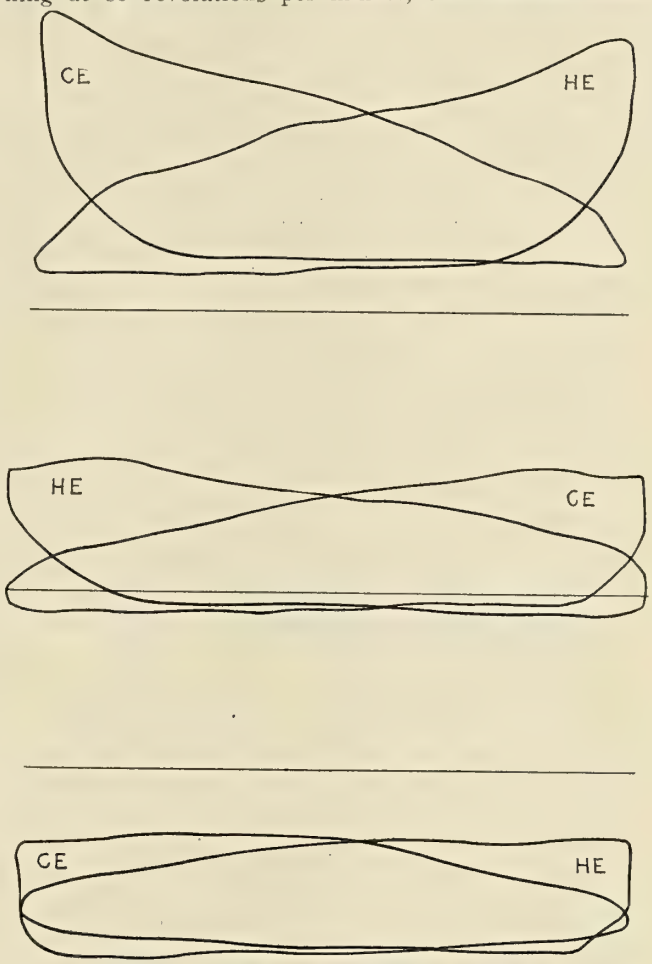
The test data were obtained during a voyage between the ports of Savannah, Ga., and New York City and return, from April 14 to 20, 1914. Thompson indicators were attached to the cylinders and cards taken from each end of each cylinder every half hour during six hours each day. Each time a card was taken, readings were made of the revolutions per minute, pressures of the boiler, intermediate-pressure and low-pressure receivers, vacuum, and the distance the ship had gone by the log during that period. From these data calculations were made of the mean effective pressure, indicated horsepower, end thrust and steam consumption.

The set of indicator cards reproduced on this page were taken with a boiler pressure of 152 pounds gage and an intermediate receiver pressure of 23 pounds gage, and a low-pressure receiver of 1.5 pounds gage. The vacuum was 28 inches, the barometer reading 29.7 inches, and the engine was turning at the rate of 67.3 revolutions per minute.

Taking first the high-pressure card, the mean effective pressure of the head end is found to be 61.3 pounds and the indicated horsepower 265.48. The mean effective pressure of the crank end of the high-pressure cylinder is 65 pounds, and the indicated horsepower 266.51. Next, taking the intermediate-pressure cylinder, the mean effective pressure of the head end is 18 pounds and the indicator horsepower 213.21. The mean effective pressure of the crank end is 17.6 pounds and the indicated horsepower 204.41. From the low-pressure card it is found that the mean effective pressure of the head end is 6.4 pounds and the indicated horsepower 212.59, while the mean effective pressure of the crank end is 6.8 pounds and the indicated horsepower 224.31.

The total indicated horsepower developed by the engine at 67.3 revolutions per minute, as computed from these cards is, therefore, 1,386.51.

From the observed data recorded the lowest number of revolutions was 61.8 and the highest 80. At 61.8 revolutions per minute the engine developed 1,219.57 indicated horsepower. The pounds of steam used per hour was 15,276.96, the number of pounds steam used per revolution 4.12, and the number of pounds of steam used per indicated horsepower per hour 12.5. With the engine running at 80 revolutions per minute, the indicated horsepower developed was 2,843.95, the number of pounds of steam used per hour 30,816, the number of pounds used per revolution 6.42, and the number of pounds steam used per indicated horsepower per hour 10.8. The propeller thrust varied from 16.82 tons with the engine running at 61.8 revolutions per minute to 35.55 tons with the engine running at 80 revolutions per minute.



Set of Indicator Cards from Main Engine of *City of St. Louis*

In making the trip from Savannah to New York, the engines were run at slower speed, the boiler pressure averaged 150 pounds, and the point of cut-off in the high-pressure cylinder was made about .5 of the stroke, while on the return trip the engines ran faster, the boiler pressure was higher, and the cut-off lengthened out to about .7 of the stroke. In consequence of which we made the return trip in several hours' less time. I was told that the reason for varying the speed in the two trips is due to the fact that the boat can put into port at New York only at specified times, whereas they can go into Savannah at any time. The higher speeds are accomplished at

the expense of the boilers and a greater coal consumption, but upon the assumption that the cost of propelling this steamer varies as the cube of her speed, and that the overhead expenses are $100 (20/16/8) per hour, we have worked out a problem to show the most economical speed, under the conditions, at which she could be run.

When the speed is 10 miles per hour, the cost is $25 (5/4/2) per hour. Therefore let

Let C = cost
K = constant
X = her most economical speed
M = the distance traveled
$C \propto X^3$
$C = KX^3$
$25 = (10)^3 K$

$$K = \frac{1}{40}$$

$$C = \frac{M}{X} \cdot \frac{X^3}{40} + \frac{M}{X} \cdot 100 = M\left[\frac{X^2}{40} + \frac{100}{X}\right]$$

Differentiating, we have,

$$\frac{DC}{DX} = \frac{2X}{40} - \frac{100}{X^2} = 0$$

Equating to zero and solving for X, gives

$$2X^3 = 4{,}000$$
$$X = 12.6 \text{ miles per hour.}$$

Discussion

The cards from this engine show it to be working under very good conditions. The work on the head and crank ends of the cylinders is well proportioned, while the same may be said of the work done by each cylinder. The steam consumption per indicated horsepower per hour averages 11.9 pounds, which speaks well for its economy. Its ideal thermal efficiency is 32.1 percent, which is high for an engine subjected to varying loads. The actual thermal efficiency is 21 percent, which, too, is all that could be expected of it.

There is but one defect so far as I could discover, and it is so slight that I doubt if it materially affects its economy. The lead on the head end of the intermediate-pressure cylinder is a little early. This is shown on the card by a little loop on the admission corner of diagram.

$\frac{T' - T''}{T'}$ is the ideal thermal efficiency of an engine expanding to back pressure. Where T' is the absolute temperature corresponding to the pressure at which it receives steam, T'' is absolute temperature corresponding to the pressure at which the steam leaves the cylinder (back pressure).

Temperature corresponding to 150 pounds gage + 461 = 826.9 degrees absolute.
Temperature corresponding to 175 pounds gage + 461 = 838.4 degrees absolute.
Temperature corresponding to vacuum 27.5 inches = 569 degrees absolute.

$$\text{Maximum ideal efficiency} = \frac{826.9 - 569}{826.9} = 31.2 \text{ percent at 150 pounds gage.}$$

$$\frac{838.4 - 569}{838.4} = 32 \text{ percent at 175 pounds gage.}$$

Actual Thermal Efficiency

Total heat in steam at 150 pounds gage from 32 degrees F. = 1,193.6 B. T. U.
Total heat in steam at 175 pounds gage from 32 degrees F. = 1,197.1 B. T. U.
Consumption in pounds per indicated horsepower per hour = 11.9 pounds.
Feed water heated to 210 degrees F.
$210 \times 32 = 178$ B. T. U. in feed water.
1,193.6 — 178 = 1,015.6 B. T. U. heat actually in the steam.
1,197.6 — 178 = 1,019.1 B. T. U. heat actually in the steam.
One horsepower = 33,000 foot pounds per minute.
One horsepower uses $1{,}015.6 \times 11.9 = 121{,}000$ B. T. U. per hour.

$$\frac{990{,}000 \times 60}{778} = 2{,}545 \text{ B. T. U. per hour.}$$

$$\frac{2{,}545}{121{,}000} = 21 \text{ percent efficiency at 150 pounds gage.}$$

ALFRED CHINA DE LORME.

Removing a Piston from its Rod

A certain lake steamer which had been in service several years was docked in the fall for general overhauling. One of the jobs that seemed necessary was to re-turn the piston rod, which had become scored badly from the use of poor grades of packing. The intention was to turn it down smooth in a lathe and fit a good type of metallic packing.

The cylinder head was removed, the junk ring and packing rings removed and the nut on the end of the rod backed off, but when it came to remove the piston from the rod with a good sized chain hoist it was found impossible to separate them.

The piston rod was coned 1 to 6 at the top end and 1 to 3½ at the bottom, so it was an easy matter to remove the rod from the crosshead, as it did not stick tightly, and so both rod and piston were taken to the shop together. But the lathe would not take the rod with so large a piston on it, so one of the shop mechanics suggested the following simple scheme.

A piece of pipe whose inside diameter was greater than the shaft's diameter was obtained and cut off in a length equal to the distance between the piston and the crosshead nut when partially hove-up. This was slipped on over the piston rod, then an old flange for a washer, and then the crosshead nut was screwed up, thus pressing the pipe tight against the piston. The pipe was then heated by means of a blow torch, and the expansion of the material due to the heating created enough pressure to break the grip between the rod and piston, whence they could be easily removed. N.

Opportunity for Export of American Machinery and Supplies to Russia

The Russian Society for Electrical Enterprises, Ltd., 5 Marsowo Polje, Petrograd, Russia, writes to INTERNATIONAL MARINE ENGINEERING to state that the Society is opening branches in various parts of the Russian Empire, and that it wishes to make arrangements to represent American manufacturers of electrical machines and appliances of all kinds, steam engines and steam specialties, internal combustion engines, etc. Catalogues and other information should be sent direct to the Society.

LAUNCH OF LAKE OIL TANKER.—The Collingwood Shipbuilding Company, Ltd., Collingwood, Ont., recently launched the steamer *Royalite* for the Imperial Oil Company, Ltd. The vessel is the first of three sister ships being built by the Collingwood Shipbuilding Company for the Imperial Oil Company. They will be employed in the oil-carrying trade on the lakes, but are to have all necessary equipment for ocean service. The steamer's dimensions are 258 feet overall, 43 feet beam and 18 feet molded depth. Her equipment is of the most up-to-date description.

Questions and Answers for Marine Engineers

Inquiries of General Interest Regarding Marine Engineering and Shipbuilding will be Answered in this Department

CONDUCTED BY H. A. EVERETT *

This department is maintained for the service of practical marine engineers, draftsmen and shipbuilders. All inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given permission to do so. Indicator cards taken from marine engines will be carefully analyzed, the defects pointed out, and the horsepower calculated, provided complete data are sent with the cards.

Method of Testing a Gasoline Petrol Engine

Q.—When a gasoline (petrol) engine is tested on the blocks previous to its acceptance, is there any customary procedure followed in making the tests—i. e., there are tests run at varying revolutions per minute and load, and what I want to know is how these varying conditions are arrived at. There seem so many variables, as spark, throttle, brake, etc., that some are undoubtedly kept in a constant given condition. If you would outline the steps for a complete test I should greatly appreciate it. G. E.

A.—Gasoline (petrol) engines, in general, operate under two conditions of loading: (A) When the speed is maintained constant and the load may vary, as is the case when connected to electric generators, and (B) they may operate under conditions of varying speed and varying load when the load is a function of the speed, as is the case when installed in a motor boat. Complete tests should investigate both conditions, and the best procedure is as follows:

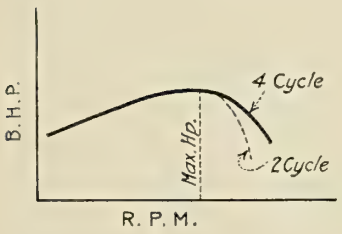


Fig. 1

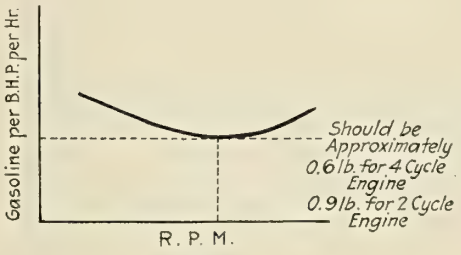


Fig. 2

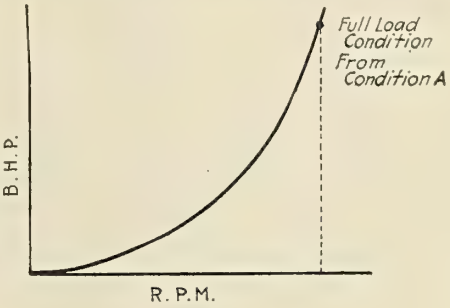


Fig. 3

Condition A. (Constant speed.) To determine the best speed, operate the engine with wide open throttle and best spark at all speeds up to no load or racing speed, and obtain curve of Fig. 1. For the individual runs set the throttle wide open and adjust the spark at each speed. Obtain speed variation by varying the brake load. At the same time derive the fuel consumption curve as shown in Fig. 2. The maximum and minimum of these curves occur at the revolutions per minute which are the best revolutions per minute for the engine to operate at when operating at full power.

Passing now to condition B, when the engine is to be used in a motor boat. Decide from the above curves the best revolutions per minute and select or design a propeller to hold the motor to not more than this speed with wide open throttle, then plot the curve of probable horsepower on revolutions per minute for the hull. This curve will be of the order H. P. = K $\times$ (R. P. M.)^3, so that if the full load condition be taken from (A) condition runs, the other points can be determined by making the horsepower vary as the cube of the R. P. M.—that is, at ½ R. P. M. (which is nearly ½ speed) the horsepower would be $(\frac{1}{2})^3 = \frac{1}{8}$ of the maximum.

Then in testing the motor at fractional speeds, select the horsepower and revolutions per minute from this curve. Get the speed (R. P. M.) by means of the throttle and adjust the brake to give the horsepower in accordance with the curve of Fig. 3.

Highest Steam Pressure Recorded—Velocity of Flow of Steam

Q.—Will you please answer the following questions in your next issue: (1) What is the highest steam pressure that can be generated? (2) What is the velocity of steam at one pound gage pressure flowing into a vacuum of 29 inches? What is it at 200 pounds gage flowing into the atmosphere, and how may it be calculated? N. P.

A.—(1) The highest steam pressure which has been reported is a pressure of 3,200 pounds per square inch at a temperature of 706.1 degrees F. This was reported by Holborn and Bauman in 1910 (*Annalen der Physik*, 1910) and is given by them as the pressure and temperature corresponding to the critical state of saturated steam. This has been accepted by authorities as being reliable, and is probably the highest pressure which has been actually obtained in a vessel containing steam.

(2) Adiabatic expansion of saturated steam in a properly designed nozzle (assuming no frictional losses) from 1 pound gage to 29 inches vacuum would give a velocity of 3,240 feet per second, assuming the barometer to be at 30 inches. Similar expansion from 200 pounds gage to atmosphere would give 3,120 feet per second.

The calculations are made as follows:

$$V = 224\sqrt{H_1 - H_2}$$

where V = velocity in feet per second
H_1 = heat contents of steam at upper pressure
H_2 = heat contents of steam at lower pressure

The heat contents H_1 for the first case is found from any good temperature entropy diagram or table as 1,150 B. T. U. That for the lower pressure and same entropy is 943 B. T. U. Therefore

$V = 224\sqrt{1{,}150 - 943} = 224 \times 14.48 = 3{,}240$ feet per second.

And similarly for the second case

$H_1 = 1{,}197$
$H_2 = 1{,}003$
$H_1 - H_2 = 194$
$V = 224 \times 13.93 = 3{,}120$ feet per second.

Calculations of this sort are best made by aid of the Mollier diagram, which is a diagram of total heats (heat

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contents) plotted on entropies. This diagram accompanies most of the modern tables of properties of saturated steam. In the above formulas, $224 = \sqrt{2g \times 778}$.

Number of "Live" or Effective Coils in a Spring

Q.—I shall now be exceedingly obliged if you will show by diagram the number of "live" or effective coils in any spring; how to count them correctly. X.

A.—Begin at that part of the coil which is unreduced in section, as *A*, Fig. 1, jump to the corresponding part of the next coil, as *B*, and so on successively down the spring till there is but a fraction of a complete jump remaining,

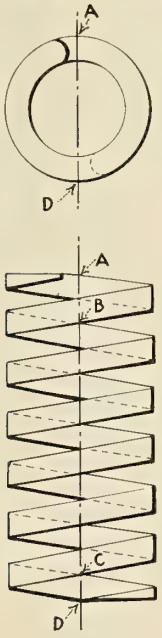


Fig. 1

as *C* in Fig. 1. Then estimate the fractional part of a complete coil remaining, as *C-D* and add it to the complete coils counted. If the ends of the spring are of less pitch as well as tapered, judgment is your only guide as to how much of a coil they are to count for. Obviously, if any two coils touch, all coils from the point of contact on are dead and should not enter into the calculation.

Percentage of CO_2 in Flue Gases for Good Operation

Q.—What should be the percent of CO_2 in flue gases for good operation at sea? Is 9 percent good, and if not can you tell me whether the efficiency of the boiler would be improved if I had a larger percentage and about how much? Fireman.

A.—Nine percent is lower than should be obtained with good fuel and firing. The values commonly met with range from 8 percent, which is considered poor, to 12 percent, which is considered good. If with 12 percent CO_2 the boiler efficiency is 67 percent, it will be about *

66½ percent for 11 percent CO_2
66 percent for 10 percent CO_2
64 percent for 9 percent CO_2
60 percent for 8 percent CO_2

It is obvious that increasing CO_2 from 11 percent to 12 percent effects much less gain in efficiency than increasing it from 8 percent to 9 percent.

* Lucke, "Handbook of Thermodynamics."

Shipbuilding Boom in China.—Hong Kong is enjoying a shipbuilding boom on an extensive and unprecedented scale. At the Taikoo yard every slip is full, the work on hand consisting of six new ships. The Taikoo yard is now so equipped that it is able to deal with all classes of vessels up to 10,000 tons, and, in view of the conditions existing in Europe, the company expects many more orders for new ships.

NEW BOOKS FOR MARINE ENGINEERS

Design of Steam Boilers and Pressure Vessels. By George B. Haven, S. B., and George W. Swett, S. B. Size, 5¾ by 9 inches. Pages, 416. Illustrations, 197. New York, 1915: John Wiley & Sons, Inc. Price, $3 net.

In writing this book, the aim of the authors, who are professors of machine design at the Massachusetts Institute of Technology, has been, as stated in the preface, to give the most practical procedure possible in the design of boilers and other pressure apparatus.

Wherever possible results have been obtained by rational rather than empirical methods. The usages of current boiler-making practice have been kept constantly in view. In the application of the principles and formulæ deduced in the early chapters of the book to the practical design of various types of boilers and pressure vessels, the data and constants assumed are derived from actual practice and represent wide margins of safety in regard to boiler performance and construction.

In discussing the general principles of boiler design, the authors take up types of boilers, circulation, materials, the factor of safety, thermal efficiency, boiler horsepower, size of units, range of pressure, superheaters, fuel, grates and settings, and a summary is given which well represents the requirements of modern practice.

Chapter II contains a mathematical discussion of the stresses in pressure vessels, and gives practical rules and formulæ by means of which such problems are solved in actual practice. Chapter III is a very complete discussion of the subject of riveted joints and is splendidly illustrated, not only by diagrams of the various joints with an explanation of the calculations for efficiency and strength, but also by numerous diagrams and tables which give the volume of rivet heads, the allowance for rivet heads, rivet strength and maximum value of pitch and efficiency of various joints. Chapter IV is on general proportions of boilers, and here the data are based on actual practice and are presented in most convenient form, frequently by tables and plots.

The remaining five chapters illustrate the application of the principles and formulæ to the practical design of various types of boilers and pressure vessels. Chapters V, VI, VII and VIII give complete specifications and calculations for the design of a horizontal return tubular boiler, a dry back Scotch boiler with combustion chamber in shell extension, a vertical straight shell multitubular boiler and a locomotive type boiler for contractors' use. The final chapter is on tank design. To illustrate the problems in design, large plates are shown, giving the complete drawings.

How to Save a Big Ship From Sinking. By Charles V. A. Eley. Size, 5 by 7¼ inches. Pages, 195. Numerous illustrations. London, 1915. Simpkin, Marshall, Hamilton, Kent & Co., Ltd. Price, 3/6 net.

Dealing with the salving of a vessel, either in preventing its sinking or after it has sunk by collision or mine or torpedo attack, as a strictly engineering problem, the author suggests an installation of salvage pumps capable of utilizing the whole of the power existing in the ship for propulsion for expelling water from the vessel. The best kind of pump to use for the work he considers would be the involute centrifugal single stage type, although in certain cases the expulsor form of steam pump, or even steam jet pumps, might be considered. Estimates of the probable additional weight and cost of such installations point, in the author's opinion, to the feasibility of such a system.

Marine Articles in the Engineering Press

Heat Losses—Improved Steering Gear—Stability—New Vessels —Model Experiments—Alternative Methods of Propulsion

Heat Losses in Steam Transmission.—By W. L. Cathcart. This article discusses mathematically the transmission of heat from one body to another by radiation and conduction and gives Péclet's formulæ for heat losses due to radiation and conduction. By means of the formulæ thus reviewed, the heat losses of uncovered and covered pipes are compared for both saturated and superheated steam. 2 illustrations. 10,000 words.—*Journal of the American Society of Naval Engineers,* August.

Improvement of Screw Steering Gear.—The author in the comparison of a usual screw steering gear with his proposed reversed tiller gear points out how the first in its way from center position to hard over has a more uniform angular velocity, a smaller moment up to the end and a more steadily increasing power demand than the proposed arrangement which in the hard over position has a very large effective leverage with decreased angular velocity and a small power demand. The author comes to the conclusion that a smaller steering engine could furnish improved steering capacity in the proposed arrangement. The article takes note also of the German experiments on the greatly increased rudder pressures, when the rudder is laid from hard over to hard over. 7 illustrations. 3,400 words.—*Schiffbau,* October 27.

A Method of Estimating the Stability Required by a Ship.—By Arthur R. Liddell. Owing to the fact that there is a current belief that the stability of a ship is an intangible quality that is in itself a mystery, the author attempts to show a matter-of-fact way of dealing with the subject so that various factors affecting stability will be more clearly understood. A table is given which embodies a method of quickly obtaining the metacentric height corresponding with an ordinate of the curve of moments at 30 degrees equal to zero. To obtain the metacentric height corresponding with the ordinate of the curve of moments for the inclination of 30 degrees, the decimal value from the table is multiplied by the breadth of the ship and the result is then added to twice the length of the lever corresponding with the ordinate of the moment curve at 30 degrees. To enable stability to be properly dealt with, the author maintains that some recognized authority should take it up. A committee or central office should be formed, the functions of which should be the collection of material, the analyzing of cases of accident and the gradual establishment of a body of natural laws and conditions on the basis of which professional judgment could be confidently founded. In view of the mass of knowledge thus accumulated, stability would cease to be the mystery that it now is. 1 illustration. 2,200 words.—*The Engineer,* November 5.

Torpedo Boat Destroyer Ericcson.—By W. H. A. Lange. The United States destroyer *Ericcson,* authorized by Congress August 22, 1912, was built by the New York Shipbuilding Company, Camden, N. J., to the following dimensions: Length overall, 305 feet 3 inches; beam on load waterline, 30 feet 6 inches; mean draft, 9 feet 9 inches; normal displacement, 1,090 tons. Propulsion is by two shafts actuated by turbines of 17,000 shaft horsepower, which at 600 revolutions per minute drive the ship at a speed of 29 knots. The machinery arrangement is starboard shaft, main high-pressure turbine, independent astern turbine; port shaft, low-pressure turbine combined with astern turbine. Coupled forward of the low-pressure turbine is a vertical direct-acting compound reciprocating engine, 16 by 24 by 18 inches stroke connected to the turbine shaft by a Metten hydraulic clutch. The reciprocating engine is used for cruising speeds up to 12½ or 16 knots. Steam is supplied at 265 pounds pressure by four Thornycroft oil-burning boilers with a total heating surface of 23,936. The propellers are three-bladed, 91½ inches diameter by 78 inches pitch with a projected area of each propeller of 4,024 square inches and a developed area of 4,500 square inches. On the standardization trials a speed of 29.33 knots was attained with the propellers turning at 616.98 revolutions per minute and the engines developing 17,796 shaft horsepower. On the four-hour full power trial the speed was 29.29 knots and the water consumption of all machinery per shaft horsepower main engines was 15.762 pounds. The fuel oil consumption was 1.23 pounds per shaft horsepower per hour and the water evaporated per pound of oil per hour was 12.818 pounds. 3 illustrations. 3,400 words.—*Journal of the American Society of Naval Engineers,* August.

The Cylindrical Tank Oil-Carrying Steamers Breifond and Aztec.—One of the recent developments of a more or less important nature in the construction of vessels for carrying oil in bulk at sea is the method employed in the cylindrical tank oil-carrying steamers *Breifond* and *Aztec.* On account of the fact that a cylindrical form will withstand more pressure than a form composed of flat surfaces of equal scantling, this fundamental principle has been adapted to the requirements of the bulk-oil trade in these two vessels by fitting the carrying space with large cylindrical retainers in which the oil is carried, instead of in ordinary holds with straight bulkheads. Both vessels were built and engined by Messrs. William Doxford & Sons, of Sunderland. They are sister ships of the following dimensions: Length overall, 400 feet; length between perpendiculars, 390 feet; breadth extreme, 53 feet; depth, molded, 32 feet 9 inches; draft, load, 25 feet 6 inches; gross tonnage, 5,711; net tonnage, 2,353; deadweight capacity, tons, 8,500; displacement, tons, 11,550. One continuous deck extends the full length of the ship. The forepeak is constructed for the storage of oils, and immediately aft of the forepeak are fitted six cylindrical tanks, above which is an expansion trunk. Light, refined oils may be carried in the circular tanks, the double bottom and the spaces outside the cylinders, but, in order to facilitate cleaning, the dirty oils are carried in the cylinders only. Space between cylindrical tanks Nos. 4 and 5 is utilized for the cargo pump room, while aft of No. 6 tank is a cross bunker hold. The crew is quartered in the forecastle, the captain and officers in a deckhouse amidships, and the engineers in the poop. Propulsion is by a single set of triple-expansion engines with cylinders 21½, 44 and 74 inches diameter by 48 inches stroke, supplied with steam by two four-furnace Scotch boilers working under Howden's forced-draft and arranged for burning either oil or coal. The machinery is situated aft. The pumping plant consists of two large duplex horizontal pumps, each capable of discharging 400 tons of oil per hour. Connection is made to all the tanks of the ship, including the double bottom, the same machinery dis-

charging the water ballast, as well as the oil cargo. 6 illustrations. 2,000 words.—*The Shipbuilder,* October.

Notes on Model Experiments.—By G. S. Baker, superintendent of the William Froude National Tank. The paper gives an account of some of the work done in the William Froude National Tank during the last two or three years, showing the mode of carrying out the experiments and the application of tank work to ships and the actual reductions of power effected in a number of cases. The work is dealt with under three headings—skin friction resistance, strut and bilge keel resistance, and the effect of hollow and straight ends. In the first section a new analysis of the previous work is given, together with the results of further experiments on friction and the skin friction resistance for all lengths of planks, is reduced to a simple form and plotted as a single curve from which ship resistance can be rapidly calculated. The effect of temperature upon the frictional resistance of a model is dealt with. An account is given of some experiments to determine the increase in resistance due to plate edges and butts, and the results are compared with W. & R. E. Froude's work. The second section deals mainly with the effect of size on strut resistance, and incidentally the applicability of model appendage experiments to the full-size ship. It also deals with the effect of fore-and-aft position of bilge keel on its resistance. The third section contains an analysis of a very complete set of experiments on hollow and straight lines and shows the limits of usefulness of either form and how these limits depend upon the prismatic coefficient and the speed. 4 plates. 5 illustrations. 11,000 words.—*Transactions of the North-East Coast Institution of Engineers and Shipbuilders,* December.

Some Alternative Designs of Propelling Machinery for a 19¼-Knot Steamer.—By James Dornan. In order to compare various systems of propulsion as regards power, efficiencies, coal consumption, weights and annual savings effected, the design of a 19¼-knot steamer of 21,000 tons displacement on a draft of 27 feet was chosen and the comparisons worked out based on data from the actual performance of similar ships. The vessel chosen was 600 feet long, 72-feet beam and 46-feet depth, of 10,000 net registered tons, an intermediate passenger and cargo type usually engaged in the North Atlantic trade. Seven designs were compared, namely: Two-shaft, quadruple-expansion engines using saturated steam; four-shaft, direct turbine drive with saturated steam; two-shaft, quadruple-expansion engines with superheated steam; four-shaft, direct turbine drive with superheated steam; two-shaft hydraulic geared transmission with superheated steam; two-shaft geared turbine drive with superheated steam, and a two-shaft turbo electric geared drive. As compared with the first design (two-shaft quadruple engines with saturated steam) the various other designs showed increased economy in coal consumption varying from 5 percent for the four-shaft direct turbine drive with saturated steam up to 35 percent for the two-shaft turbo electric gear. The most important figures presented in the paper, however, are those for the net annual return on cost in percent. These were as follows: Two-shaft, quadruple-expansion engines with saturated steam, 7 percent; four-shaft, direct turbine drive with saturated steam, 9½ percent; two-shaft quadruple engines with superheated steam, 9 percent; four-shaft, direct turbine drive with superheated steam, 10 percent; two-shaft turbines with hydraulic gear, 12¼ percent; two-shaft, geared turbine drive with superheated steam, 11¾ percent, and two-shaft, turbo electric geared drive with superheated steam, 15 percent. 4 plates. 6,500 words.—*Transactions of the Institution of Engineers and Shipbuilders in Scotland,* November.

Canadian Railway Ferry Steamer Scotia II.—The *Scotia II* is a railway ferry steamer built recently by the Walker Shipyard, of Sir W. G. Armstrong, Whitworth & Company, Ltd., Newcastle-on-Tyne, to the order of the Minister of Railways and Canals of the Dominion of Canada in connection with the Intercolonial Railway System. The vessel is intended for service across the Strait of Canso, between the mainland in Nova Scotia and Cape Breton. The hull is 300 feet long overall, 50 feet beam extreme, 20-feet depth molded, and at a draft of 14 feet 2 inches displaces 3,665 tons. The vessel is of the single-deck type, with three railway tracks on the main deck. The hull is strengthened for ice-breaking purposes, and also for this purpose the propelling machinery was designed in two separate units on the center fore-and-aft line, the forward engines working the bow propeller and the after engines the stern propeller. A suitable coupling is fitted so that the two engines may work together or independently. Each set of engines is of the triple-expansion type, with cylinders 21, 34 and 54 inches diameter, with a stroke of 30 inches. Steam is supplied at 160 pounds pressure by four single-ended Scotch boilers arranged in the wings of the ship, with uptakes leading to funnels at each side. All of the auxiliaries connected with the propelling machinery are located in the port wing, while in the starboard wing are the electric generating sets, air pumps and air receivers for ship service and for engineers' stores. The machinery is designed to indicate 3,000 horsepower. 20 illustrations. 1,500 words. —*Engineering,* October 20 and November 19.

The Western Australian Government Motor-Ship Kangaroo.—The motor-ship *Kangaroo,* acquired by the Western Australian Government to carry produce from Western Australia to England, a voyage through the Suez Canal and back around the Cape of over 20,000 miles, was built by Messrs. Harland & Wolff, Ltd., Belfast, and equipped with Diesel type engines by the Burmeister & Wain Oil Engine Company, Ltd., Glasgow. She is 365 feet long between perpendiculars, 50 feet molded beam and 29 feet molded depth to the upper deck, with a gross tonnage of 4,348, a net tonnage of 2,777, and at a load draft of 23 feet 4½ inches, has a displacement of 6,640 tons. The bunker oil tanks have a total capacity of 814 tons, and the water ballast tanks a total capacity of 872 tons. Six transverse watertight bulkheads are fitted, and the upper and main decks are plated all fore and aft except in way of the hatches. Accommodations are provided for a number of first-class passengers. Two 100-kilowatt dynamos are fitted, driven by the Diesel oil engines that also drive the compressors in connection with the main motors. All the machinery, including the auxiliaries from the motor room, is electrically driven. 5 illustrations. 1,300 words.—*Engineering,* November 5.

Survey of New Jersey Ship Canal.—The proposed New Jersey ship canal, which is to run between the Delaware River and Raritan Bay, is to be surveyed during the winter under authority from the State of New Jersey. The surveys will be in charge of W. F. K. Griffith, under the direction of B. F. Cresson, Jr., chief engineer of the Department of Commerce and Navigation of the State. The route for this canal has already been surveyed from Bordentown to Jamesburg, so that the work remaining is for the section between Jamesburg and Morgan, the latter being the terminal point in Raritan Bay.

Shipbuilding and General Marine News

Contracts for New Ships—Marine Terminal Improvements—Recent Launchings — Improved Appliances — Personal Items

Several events of importance to marine interests have occurred sinced the first of the year, apart from the placing of a large number of contracts for new vessels and other marine construction work.

The Long Beach shipyard, Long Beach, Cal., has been taken over by the California Shipbuilding Company, primarily for the construction of submarines for the United States navy, although facilities are available at this yard for the construction of merchant vessels and it is reported that the yard will soon begin work on a large freighter.

Of special interest to shipowners is the recent announcement that the New York Shipbuilding Company, Camden, N. J., and the Newport News Shipbuilding and Dry Dock Company, Newport News, Va., have signed contracts with the Werkspoor Works, Amsterdam, Holland, for the rights to construct in this country Werkspoor marine Diesel engines for merchant and naval vessels. This marks an important step in the development of marine Diesel engine practice in the United States, as this type of engine has proved eminently suitable for economical operation in large ocean-going ships.

Shipbuilders on the sea coast and Great Lakes have recently placed orders for over 50,000 tons of plates, bars, shapes and forgings for delivery during the second and third quarters of the current year to cover the construction of vessels contracted for since the first of the year. Prospective contracts for ocean-going vessels will call for as much more steel during the current month, if contracts now under negotiation are closed.

Contracts for New Ships

The Fore River Shipbuilding Corporation, Quincy, Mass., has received a contract from the Government of the Argentine Republic to build a fuel steamship, 330 feet long, to cost about $600,000 (£123,000).

The Harlan & Hollingsworth Corporation, Wilmington, Del., has received a contract from the Ocean Steamship Company of Savannah (Savannah Line), New York, to build a large freight and passenger steamer. This is to be a duplicate of the ship recently ordered, which was reported in our January issue.

The W. & A. Fletcher Company, Hoboken, N. J., has received a contract from W. D. Pollock (Produce Exchange, New York), Manager Marine Department, New York Central & Hudson River Railroad Company, to build a double deck ferry-boat, also a tug boat.

The New York Shipbuilding Company, Camden, N. J., will build six steel car floats for the New York Central & Hudson River Railroad Company.

The Bath Iron Works, Bath, Me., will build two car floats, each 205 feet long.

The Baltimore Dry Docks & Shipbuilding Company, Baltimore, Md., will convert the British ship *Calgary* into an oil tanker at a cost price said to be $75,000 (£15,380).

Harry Cossey, Tottenville, S. I., is building fifty wooden vessels as follows: ten steam derrick lighters, ten lumber barges, fifteen hand winch lighters, ten gasoline derrick lighters and five 1,400-ton grain barges.

It is reported that Sawyer Brothers, Millbridge, Me., will build a three-masted schooner on speculation.

R. L. Bean, Camden, N. J., will build a four-masted schooner of 1,500 tons carrying capacity.

It is reported that the Standard Shipbuilding Corporation, Shooters Island, N. J., has received contracts for all the ships that can be constructed at this yard during the next two years. All these ships are stated to be exactly alike, being 6,000-ton freighters of standard shape.

It is reported that the Baltimore Dry Docks & Shipbuilding Company, Baltimore, Md., has received contracts from Norway to build four more oil engined ships.

The American Shipbuilding Company, Cleveland, Ohio, has received a contract from the Pittsburgh Steamship Company, Cleveland, to build another 600-foot ore boat, making three of that size ordered for 1916 delivery.

It is reported that the American Shipbuilding Company has received a contract from Carl D. Bradley, president of the Limestone Transportation Company, Rogers City, for a $500,000 (£102,500) ship for 1917 delivery.

The United Engineering Works, San Francisco, Cal., has received a contract from C. Henry Smith, Inc., San Francisco, Cal., to build a 340-foot steel freight steamship.

The Moore & Scott Iron Works, Oakland, Cal., have received a contract from Hind, Rolph & Co., San Francisco, to build a freight steamship of 7,000 tons carrying capacity, to cost about $700,000 (£143,300).

The Anderson Steamboat Company, Seattle, Wash., has received a contract from A. F. Thane & Co., San Francisco, Cal., to build four steam lumber schooners. Each of these vessels will be 248 feet long, and will have a capacity of 2,500,000 feet of lumber.

Johnson Brothers & Blanchard, Seattle, Wash., have a contract from the Westwood Navigation Company, Seattle, to build a 150-foot wooden vessel to be equipped with twin screw gas engines.

It is reported that Seattle, Wash., shipping men, headed by R. M. Calkin, traffic manager of the Chicago, Milwaukee & St. Paul Railway, are organizing a company to build and operate three steamships to ply between Seattle, Honolulu, Australia and New Zealand.

Marine Terminal Construction and Improvements

The Pennsylvania Railroad Company, Broad Street Station, Philadelphia, Pa., has asked for bids for improving and enlarging Pier 46, Philadelphia.

The Donner Steel Company, recently incorporated in New York State with $5,000,000 (£1,025,000) capital, has taken over the plant of the New York State Steel Company, Buffalo, and will make extensive improvements, including building ore docks along the Buffalo River.

The Norfolk Storage Company, Inc., recently organized at Norfolk, Va., has purchased a site of 19 acres near Norfolk and will build docks and sheds.

The Freeport Sulphur Company, Freeport, Tex., is planning to construct a dock.

It is reported that Leon Fighiera, Ocean Park, Cal., is planning to build a pleasure pier, to cost about $1,000,000 (£205,000).

J. R. Jervey, engineer, Richmond, Va., has submitted to the Council Committee on Rivers and Harbors, plans

for docks and terminals at an estimated cost of about $3,000,000 (£615,000).

The City of Hampton, Va., is considering a $90,000 (£18,450) bond issue to build docks and to make other improvements.

The Board of Trustees of Manhattan, Cal., are considering the construction of a pleasure pier at a cost of about $70,000 (£14,350).

The Santa Monica Land Company, Santa Monica, Cal., is considering the construction of a 560-foot wharf.

The Board of Estimate of the City of New York has appropriated $55,000 (£11,280) for building a pier at Coney Island. R. A. C. Smith, Pier A, North River, New York, is Dock Commissioner.

New Steamship Lines

W. A. Rousch, G. W. Hubbell and others of Macon, Ga., have incorporated the Gray Boat Line.

William Wallis and Hollis Fort, of Americus, Ga., and J. M. Sumner and E. E. Cooke, of Akron, Ohio, have organized the Flint River Navigation Company, capital $25,000 (£5,130) and plan to operate boats on the Flint and Appalachicola rivers.

Frank E. Mathews is president and general manager, W. S. Bullock is treasurer and B. J. Hunter is secretary (all of Ocala, Fla.) of the recently organized Oklawaha-Silver Springs Steamboat Line.

Shipyard Extensions and Improvements

Harold M. Sewall and others of Bath, Me., have purchased the shipbuilding plant of the New England Company, of Bath, which has been idle for more than ten years. It is reported that the yard will shortly be operated for building wooden ships.

It is reported that W. A. Boole, Jr., Oakland, Cal., is at the head of a new company which will build a large shipyard at Oakland.

Isherwood System of Ship Construction

Although the year 1915, so far as Great Britain is concerned, has not offered many facilities for the building of mercantile vessels, owing to the demands made upon shipbuilders by the Government, it is very interesting to note that this has been a record period for the "Isherwood" system, orders for more Isherwood vessels having been placed this year than in any preceding year. The total number of Isherwood vessels contracted for to date in the current year is no fewer than 135, approximating to 615,315 gross tons, which is almost equal to 1,000,000 tons in deadweight carrying capacity.

The following analysis of vessels built or being built clearly demonstrates the suitability of the Isherwood system for all types of craft, and at the same time emphasizes the predominance it has gained in the construction of certain classes of vessels:

			Gross Tons
General cargo vessels.	167	Representing about	745,763
Tankers	187	" "	1,054,783
Colliers	21	" "	130,760
Great Lake freighters.	16	" "	111,745
Passenger vessels	9	" "	27,769
Barges	43	" "	12,742
Dredgers	2	" "	760
Trawler	1	" "	570
	446	" "	2,084,892

As may be readily imagined, the advantages of the Isherwood system have not been overlooked by the Government, and orders for several tank steamers have been placed by the Admiralty, and other types of boats have been contracted for by the Indian Government. In the United States, three more large freighters for the American Lake trade have been placed on order, and to evidence the firm footing the system has gained in Japan, it might be stated that 22 vessels have recently been contracted for.

Among the most interesting vessels delivered during the period under review, mention should be made of the ore carriers *Rose Castle* and *Lord Strathcona*. These vessels, which have been specially designed for the carriage of coal and ore, are the largest single-deck vessels ever built in the United Kingdom. In the United States, the *Achilles* and *Ulysses* have been placed in commission by the Panama Canal Commissioners, for whom they were specially built. Those two vessels are also high-class colliers, and are very similar to the colliers *Orion* and *Jason*, which were designed and are operated by the United States navy department. That the Isherwood system should be selected for the construction of what are undoubtedly the six finest colliers in the world would seem to indicate that the claims which have been made for it in connection with this class of steamer are based on quite a substantial foundation.

New York and Newport News Shipbuilding Companies to Build Big Motor Ships

T. Orchard Lisle, United States representative of the Netherlands Engineering Company, of Werkspoor Works, Amsterdam, Holland, announces that the Newport News Shipbuilding Company, of Newport News, Va., and the New York Shipbuilding Company, of Camdem, N. J., have signed contracts with the above well-known Dutch firm whereby they will construct under license in the United States, Werkspoor mercantile-marine and naval, four-cycle, Diesel-type motors.

For several years both the Newport News Shipbuilding Company and the New York Shipbuilding Company have been closely watching the development and progress of the Diesel-driven motor ship, and they are to be congratulated on being the first American concerns to adopt Diesel engines of a slow-speed type suitable for large ocean-going ships. Their choice of design was influenced, of course, by the splendid results accomplished by the twenty-two Werkspoor Diesel-engined ships now in service and also by the remarkable possibilities of the Werkspoor submarine Diesel engine.

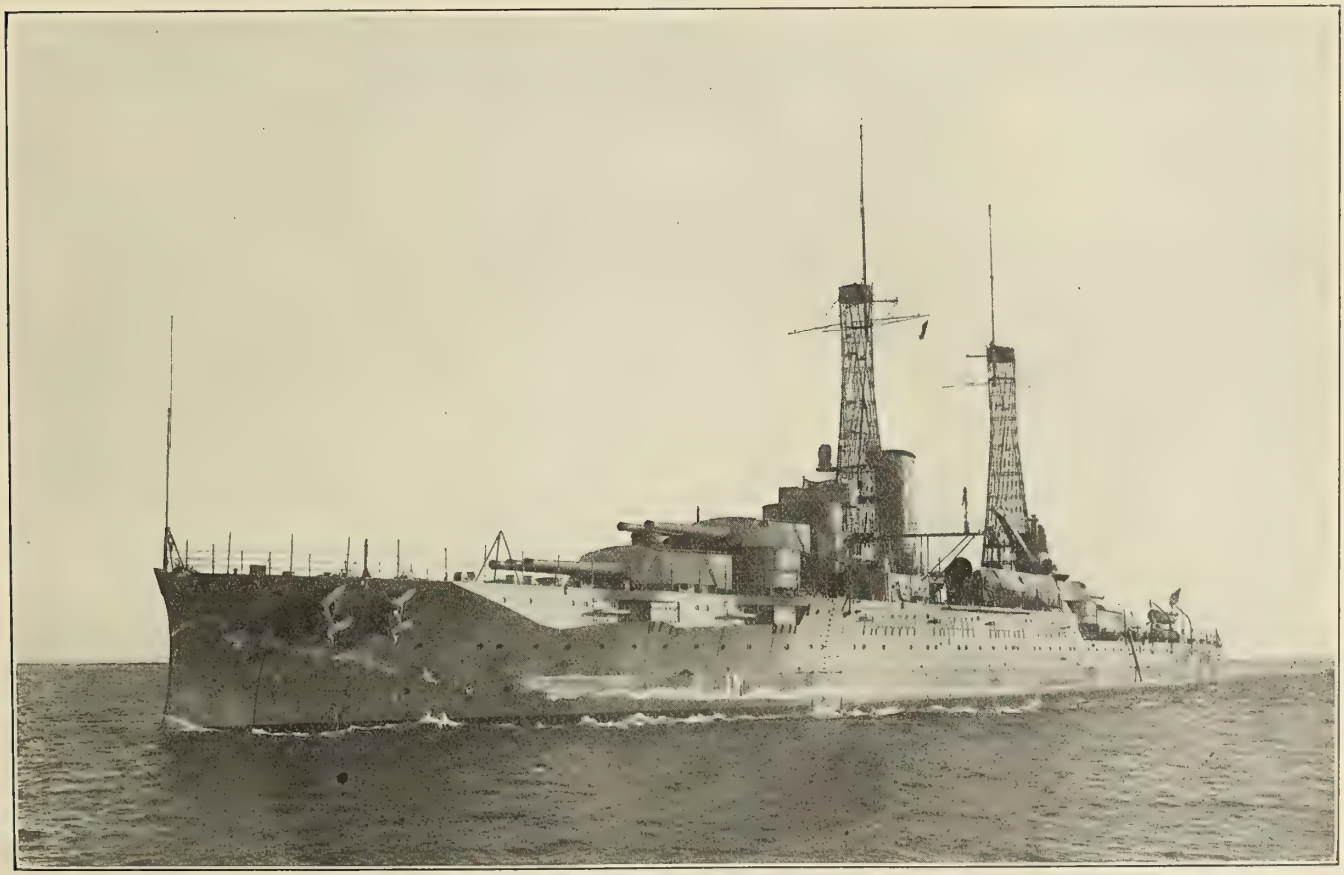
In many ways this step is important, especially when it is considered that the famous East Asiatic Company, of Copenhagen, has sold all its steamers and is rapidly replacing them with Diesel-type motor-driven liners. In fact, one well-known Danish firm of shipbuilders actually has on order more mercantile motor-driven ships than any shipbuilding firm in the world has steamships building.

The economies obtained, due to the fact that there are no large boilers or stokeholds in a motor ship, and because the consumption of oil fuel is only one-fifth that of a similar coal or oil-fired steamer, so that the bunker and boiler spaces thus saved can be given over to extra cargo, mean that with a tramp of 10,000 tons, about 600 tons of extra cargo can be carried, and at the present high rate of freights the economical advantage in a year is enormous, apart from the large saving on the fuel bill.

United States Destroyer *Jacob Jones*, Built by the New York Shipbuilding Company, Camden, N. J., on her Official Trial Trip

Oklahoma Exceeds Her Contract Speed

The new United States superdreadnought *Oklahoma,* which went through the first of her official trials over the Rockland course off the coast of Maine on January 15, attained a speed of 21.47 knots, or nearly a knot above her contract speed. The *Oklahoma,* which is driven by reciprocating engines, was built by the New York Shipbuilding Company, Camden, N. J., and is a sister ship to the *Nevada,* built by the Fore River Shipbuilding Corporation.



United States Superdreadnought *Oklahoma*, Built by the New York Shipbuilding Company. Trial Speed, 21.47 Knots

Launching of S. S. Texas

The launching of the new steamship *Texas*, the latest and finest of the Texas Company's fleet of tank steamers, took place on December 21, 1915, at the yard of the Fore River Shipbuilding Corporation, Quincy, Mass., attended by ceremonies befitting the occasion. The vessel was christened by Miss Frances Edwina Brooks.

The *Texas* went overboard in remarkably quick time, only twenty-seven seconds elapsing from the releasing of

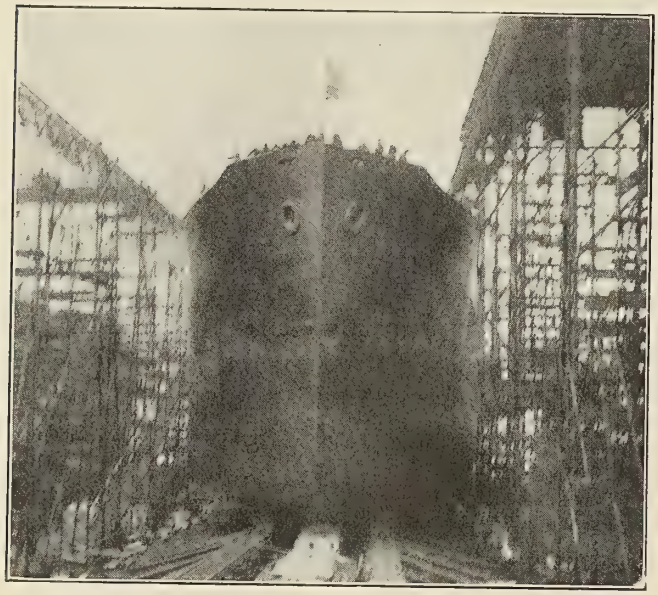


Fig. 1.—The *Texas* Going Down the Ways

the trigger until the ship was afloat. The vessel, which was designed by Mr. George B. Drake, assistant manager of the Marine Department of the Texas Company, is constructed to Lloyd's highest class, and is built on the Isherwood system of longitudinal framing. Throughout her design and construction the latest and best methods of tank-vessel construction have been followed.

The ship is 430 feet long overall, 56 feet beam, 32 feet 9 inches deep, and has two continuous steel decks, raised forecastle, bridge and poop decks, and a lower deck in the package freight hold. The hold is divided by oiltight bulkheads into compartments for sixteen main oil cargo tanks, sixteen summer tanks, fuel oil tanks, two pump rooms, cargo hold forward, machinery space aft, peak trimming tanks at ends and double bottom under engine and boiler space. The ship is schooner rigged, with two steel masts.

The *Texas* will have a total deadweight carrying capacity of 9,100 gross tons on 25-foot 6-inch draft. The cargo tanks have a total capacity of 73,000 barrels, or 3,066,000 gallons. Her contract speed is 11 knots, loaded.

Particular attention has been paid to the arrangement of quarters and everything provided for the comfort and safety of the crew, including a hospital. Marconi wireless will be used, as on other ships of the line. The *Texas* will carry various grades of refined oils without danger of mixture, and the cargo pumping system has been carefully worked out to the best point of efficiency, under the personal direction of Mr. Thompson. A refrigerating plant will be installed, complete electric lighting outfit, including searchlight, steam capstan, windlass, cargo winches, mooring machines, steering gear, etc.

The propelling machinery consists of one three-cylinder, triple-expansion engine, with cylinders 26½ inches, 44 inches and 74 inches diameter by 51 inches stroke, designed for a maximum speed of 75 revolutions per minute. Steam is supplied by three single-ended Scotch boilers, 15 feet 3 inches diameter by 11 feet long, working at 190 pounds pressure. Oil fuel will be used, with a mechanical atomizing system. A donkey boiler is also provided for use when in port and when desired to assist in getting up steam on the main boilers.

A steam heating system is provided for all living quarters, and steam heating coils for fuel tanks and such other tanks as may be used for heavy oils. Steam fire-extinguishing pipes are run to all tanks.

The *Texas* is the first of four sister ships which the Fore River Shipbuilding Corporation is building, two for the Texas Company and two for the Texas Steamship Company. The second ship, which will be named the *New York*, will be launched in about two months, and delivery of the last two vessels is called for about January and March, 1917.

The original steamship *Texas*, which has faithfully served as flagship from 1907 until the *Illinois* came out, in 1913, has now ceded both her name and former title to the new *Texas*, and will be renamed the *Georgia*.

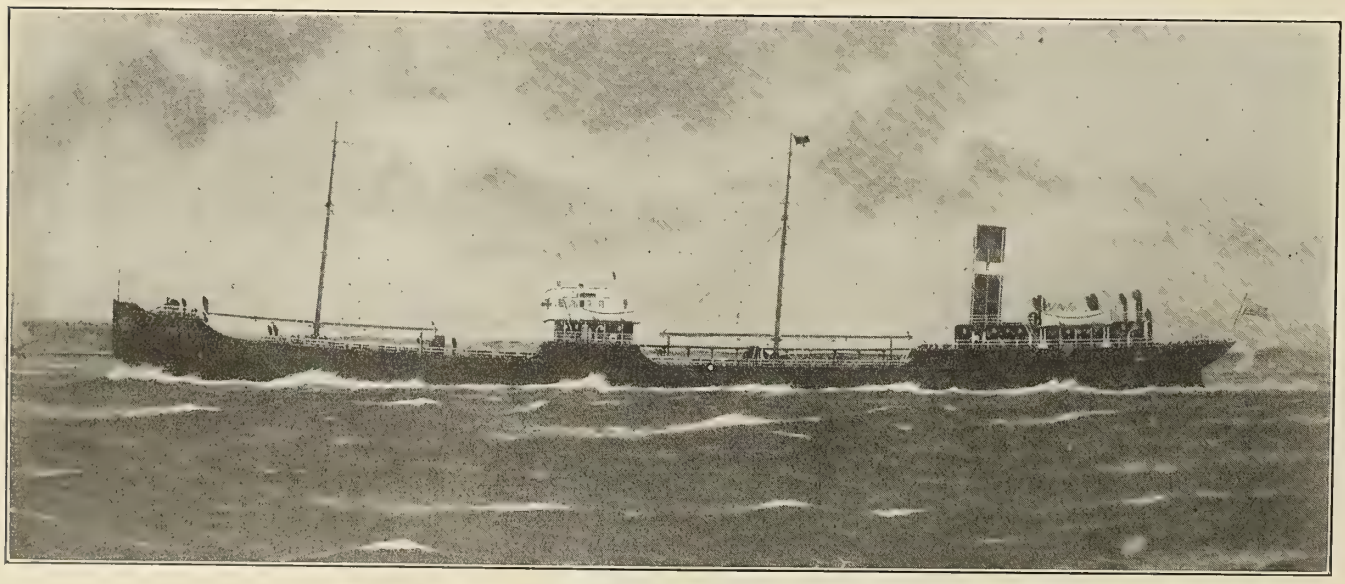


Fig. 2.—The Oil Tanker *Texas*, as she will Appear when Completed

(Photograph by Edwin Levick, New York)

A Sturdy Competitor of the Modern Steamship

ENGINEERING SPECIALTIES

Boilers of New Coastwise Colliers Equipped with Eckliff Automatic Boiler Circulators

The boilers of all of the large new colliers recently built or now under construction by the New York Shipbuilding Company, Camden, N. J., for the Coastwise Steamship Company and the Pocahontas Navigation Company have been equipped with Eckliff automatic boiler circulators, manufactured by the Eckliff Automatic Boiler Circulator Company, Detroit, Mich. In all, twenty-four Eckliff circulators have been installed in ships building for the Coastwise Transportation Company, and eight in ships for the Pocahontas Navigation Company. These vessels, which are over 300 feet long and of about 5,000 tons deadweight, capable of about 10½ knots speed at sea, form the largest fleet of colliers under American registry, and every device conducive to reliability and economy of operation has been embodied in their design. It is not surprising, therefore, that in order to improve the economy of fuel consuption and to eliminate boiler troubles due to poor circulation, Eckliff automatic boiler circulators should be installed in the boilers of all of the vessels of this fleet.

DeLaval Turbine-Driven Compressor

The construction of the DeLaval multi-stage compressor, manufactured by the DeLaval Steam Turbine Company, Trenton, N. J., is such that the casing is split horizontally, permitting of the top half or cover being lifted off, so that internal parts may be inspected and lifted out after removal of the bearing caps. The impellers are mounted upon a large-diameter and stiff shaft, the critical speed of which is far above the running speed, and which is made from hammer-forged, open-hearth steel, suitably heat treated and ground and polished over the entire surface on dead-center grinders. The hubs of the wheels and impellers are attached to the shaft by keys and separated from one another by shaft-tightening rings, which run inside the split packing rings attached to the diaphragms separating the stages. The diaphragms between stages are separate from the casing and are divided on a horizontal plane so that they may be removed without removing the shaft and impellers. When cooling is employed, the diaphragms are hollow, the water entering through the bottom of the casing and escaping at the top.

The impellers are of the single-suction, shrouded type, and are built up on heat-treated, forged, chrome-nickel steel disks, the blades or vanes being riveted to the disks and to side plates, turned tapering to give strength and lightness. That part of the disk which is within the circle of the suction opening is subjected on the inlet side to the suction pressure and on the back to the discharge pressure of the individual stage, which gives a resultant thrust equal to the area of the suction opening multiplied by the pressure generated. The accumulated thrust of all the stages is overcome by a balancing disk at the discharge end, so arranged that one side receives the total discharge pressure in a direction opposite to that acting upon the impeller disk, while the chamber upon the opposite side of the balancing disk is connected to the suction inlet, thus completely neutralizing the unbalanced pressure on all of the wheels. The head delivery characteristic can be varied considerably to meet different requirements.

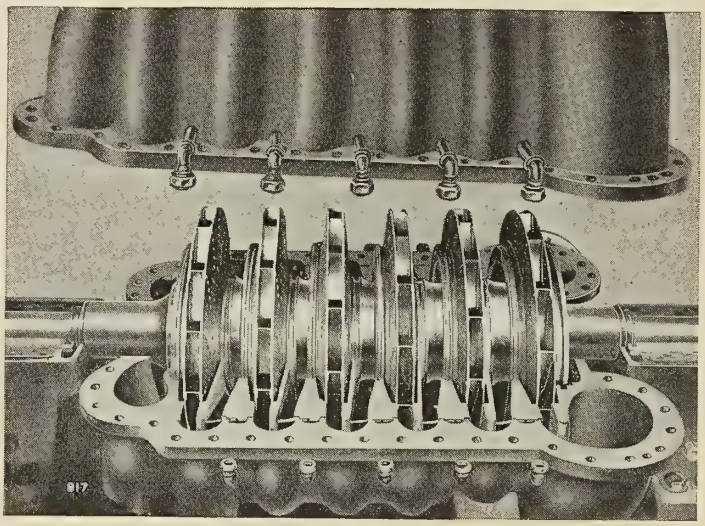


Fig. 2.—Multi-Stage Compressor with Cover Lifted

Generally where compressed air is used in tools, as around machine shops, shipyards, etc., a fairly constant pressure over a wide range of delivery is desired. The capacities in such cases range between 2,000 and 10,000 cubic feet per minute and the pressures from 75 to 120 pounds per square inch. Less than 2,000 cubic feet per minute is usually not practicable. The larger machines, or machines which must run at motor speeds, are usually built in two sections; that is, with two shafts and two housings, the speeds available with electric motors of

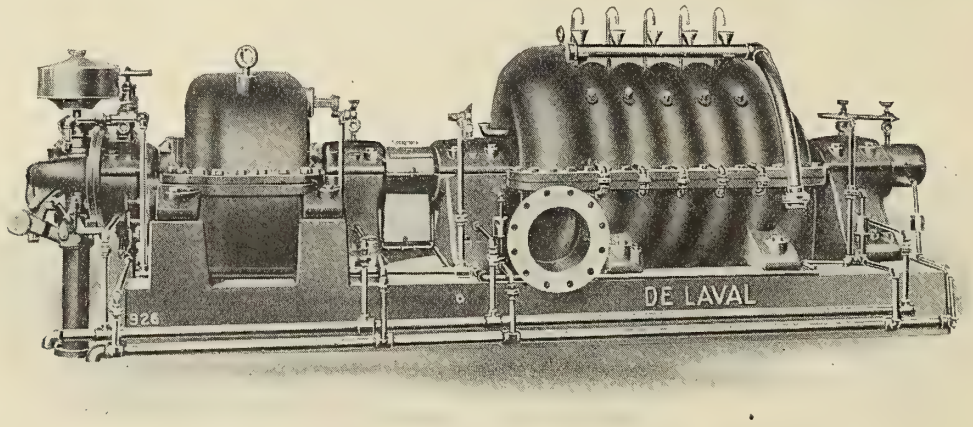


Fig. 1.—DeLaval Turbine-Driven Multi-Stage Air Compressor

these sizes requiring either a compressor with a large number of stages and several housings, or the use of a step-up gear, by means of which the compressor can be operated at 4,000 to 5,000 revolutions per minute. It may be noted that in shipyards it is frequently desirable to centralize the power generated in large turbo electric units and to operate compressors and blowers, pumps, hoists, etc., by motor, the 10 percent or so loss in electrical transmission not being so great as the loss from widely separated steam using units, and the compressor load of 1,000 or 2,000 horsepower forming a good foundation for the turbo-generator load.

The Viking Rotary Pump

A durable pump of simple construction, for which high efficiency in a large range of heads and speeds with a minimum of wear is claimed, has been placed on the market by the Schirmer-Frensdorf Company, Cleveland, Ohio, and is known as the "Viking" rotary pump. The only moving parts of the pump are a small spur gear and a large annular gear arranged as shown in Fig. 5. The small spur gear driven by the large annular gear is set off center and meshed so that its outer circle is tangent to the outer circle of the large gear at the center of the abutment between the inlet and outlet. The large gear wheel has a shaft at the back which extends through the pump casing and carries a pulley used for driving. The crescent-shaped projection of the head fills the space between the outer circle of the small gear and the inside circle of the large gear on the opposite side of the shafts.

After the water enters the pump it is carried around in the pockets formed by the spurs of both gears and the crescent-shaped abutment. When the gears begin to mesh, the water is forced into the outlet port. The abutment between the ports then prevents its passage back into the entrance port, so that it must go on into the outlet pipe.

Test runs have been made with heads varying from approximately 30 to 150 feet of water, and with speeds from 300 to 450 revolutions per minute, referring to the shaft of the large gear on which the pulley is mounted. The efficiencies were taken as the ratio of the horsepower delivered to the water pumped to the horsepower supplied to the driving pulley. The average of runs taken with a 2-inch pump showed a minimum efficiency of 48.6 percent and a maximum efficiency of 76.3 percent.

No foot valve is required for starting the water for any reasonable lift. Only one moving part requires packing, the main shaft where it enters through the back casing.

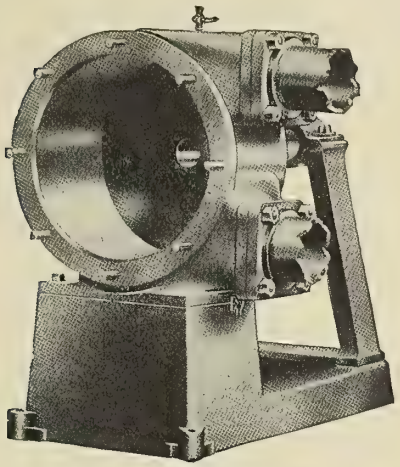


Fig. 1.—Casing of Pump

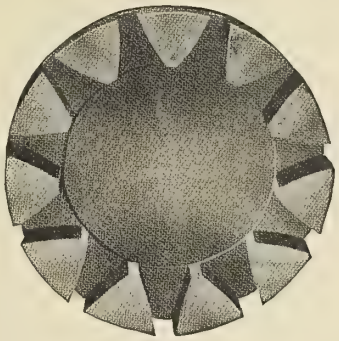


Fig. 2.—Annular Gear

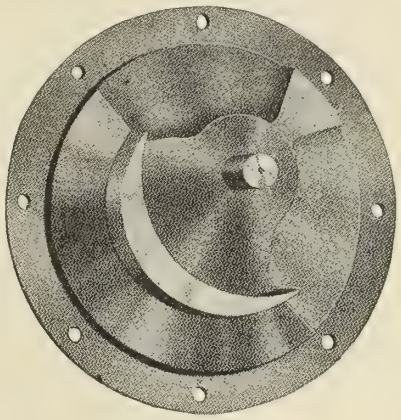


Fig. 3.—Cover

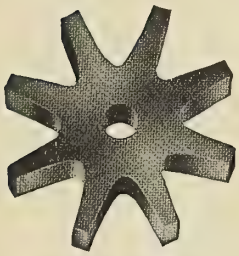


Fig. 4.—Spur Gear

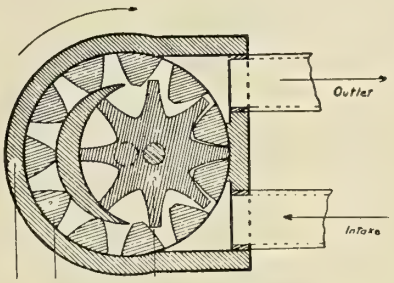
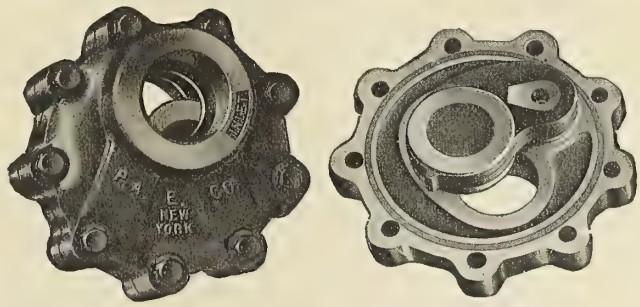


Fig. 5.—Section Through Pump

Everlasting Blow-off Valve

The Scully Steel & Iron Company, Chicago, Ill., has placed on the market a marine type of the "Everlasting" blow-off valve, which has an extra heavy body and is guaranteed to pass government inspection. The valve is especially designed to meet marine requirements and is composed of a top and bottom bonnet bolted together with an approved high-pressure gasket between them. The valve has no stuffing box, and, for this reason, requires no packing or repairing. After it is once installed it requires no attention from the engineer outside of operating it. The valve has a straight-through blow, thus eliminating the deposit of sediment in any part of the valve. The disk has a rotating movement over the valve seat, which makes it self-grinding and keeps the surface true. The inlet orifice is slightly choked, which increases the velocity of the blast at that point and insures its delivery into the discharge pipe without impairing the seat. This also has the effect of syphoning the valve clean at each operation.

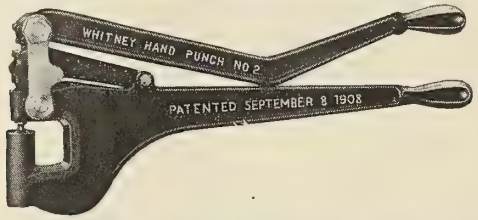


Exterior and Interior Views of Everlasting Blow-off Valve

The valve can be used anywhere where a quick opening or closing valve is required, and many large steamship companies on the Atlantic and Pacific coasts, as well as on the Great Lakes, are using the valve, not only in blow-off service, but also on purifiers. It is made with screwed or flanged ends, or with one end screwed and the other flanged, and also with angled ends. If a slow-opening valve is required it can be furnished with a rack and pinion device.

Portable Channel Iron Punch

The W. A. Whitney Manufacturing Company, of Rockford, Ill., has just added a portable channel iron punch to their line of portable hand metal punches. All parts of their No. 2 punch are interchangeable with this new channel iron punch, including punches and dies. The only



Whitney Portable Channel Iron Punch

part that is different from the No. 2 punch is the drop forged lower jaw, which has an opening back of the die socket that allows channel and angle iron to be pushed through and punched in the web between the flanges. The capacity of this channel iron punch is a ¼-inch hole through ¼-inch iron. It punches to the center of 4-inch channel iron, with 1½-inch flanges.

Cedervall Patent Protective and Lubricating Box for Propeller Shafts

To prevent the access of water to the stern tube of a vessel and to provide a reservoir of oil for lubricating the whole bearing surface of the tail shaft so that it runs virtually in an oil bath with a minimum amount of friction, F. R. Cedervall & Söner, Gothenburg, Sweden, have patented and are manufacturing a protective and adjustable lubricating box. The device, which is made of brass throughout, consists essentially of two concentrical rings fitted around the propeller shaft. The outer ring is fixed to the forward end of the propeller boss by means of screws and serves as a packing box. The inner ring, the working surface of which is faced with anti-friction metal, is pressed to the after end of the propeller tube by means of a series of spiral springs fitted between the two rings. By this arrangement it is possible to obtain a watertight joint at the stern tube. An important arrangement has also been incorporated in the lubricating box so that it is adjustable and can be lengthened or shortened as required, within practical limits and yet retain the proper compression on the spiral springs.

The protective box, together with the shaft and the end tube, forms an oil reservoir in the shape of a ring, from which the oil cannot escape, and into which the water cannot gain access. Oil is brought to the reservoir by means of grooves cut in the stern tube bush and connected with a pipe system for filling and draining the tube.

The complete arrangement of a stern tube fitted with the adjustable lubricating box and oil connections is shown in Fig 2. Three or more grooves are cut in the stern bush in the position shown in the illustration. The top one is for the escape of oil, and the others are for leading the oil to the reservoir. From the stern tube three pipes are carried through the aft bulkhead to any convenient place for connection to the supply of oil. The length of the after bush should be from four to four and one-half times the diameter of the shaft and of the forward bush from one to one and one-half times the diameter of the shaft.

The advantages claimed for this device are an increase in the number of revolutions of the engines, efficient lubrication of the tail shaft, elimination of costly brass liners and lignum vitæ bearings; elimination of galvanic action and protection of the propeller shafts from breakdowns.

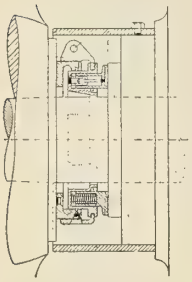


Fig. 1.—Adjustable Lubricating Box

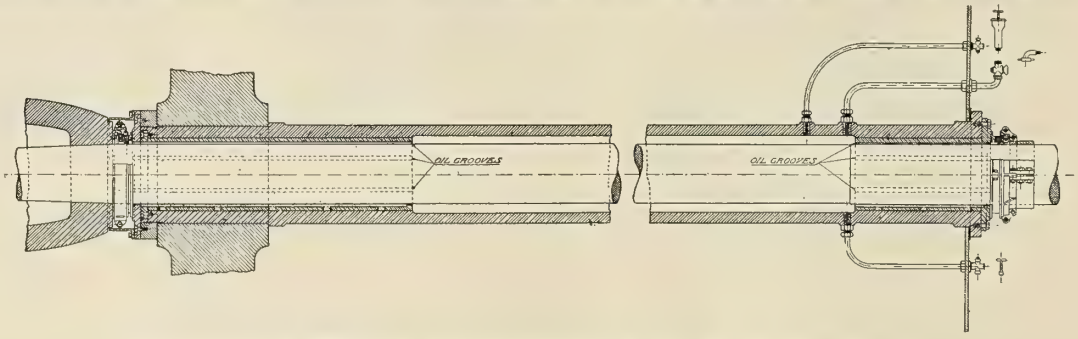
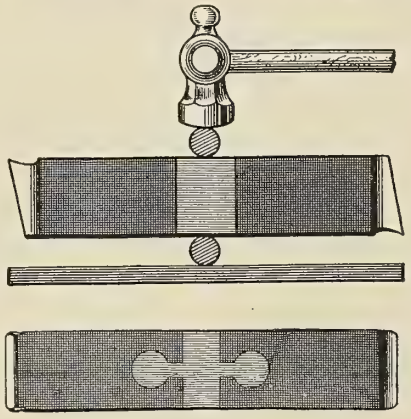


Fig. 2.—Stern Tube Fitted with Adjustable Lubricating Box and Oil Connections

"Gauge=All" Expansible Double Boring Bar

Owing to the comparatively limited quantity of high-speed tool steel which is available in the countries which are at war, and its high price and indispensability, Messrs. Vislok, Limited, 3 St. Bride's House, Salisbury Square, London, E. C., have placed on the market an expansible



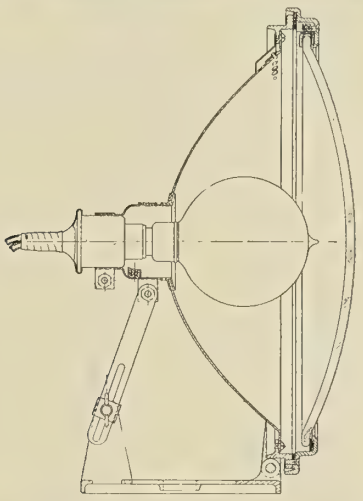
"Gauge-All" Boring Bar

double-boring cutter in which two pieces of high-speed tool steel, forming the cutters, are joined together at the center by a section of special soft metal alloy. When, after the usual wear, it is necessary to expand the tool to

its original diameter, it is only necessary to place an iron or brass clip around the center and strike it lightly with a hammer, as shown in the illustration. This compresses the metal alloy and expands the cutter. By regulating the hammer blows minutely, accurate expansion is obtained and the expansion is carried out with the cutter in use, thus eliminating frequent visits to the tool room and other delays. It is claimed that the "Gauge-All" tool enables the maximum value to be realized from the extremely expensive and increasingly rare high-speed tool steel.

Flood-Lighting Projector

A device for projecting a beam of light from a distance on to a place where work is being carried on at night, as, for instance, around docks when coaling or loading ship or on shipboard when coaling ship, has been developed by the Edison Lamp Works, General Electric Company, Harrison, N. J. The projector consists of a highly polished aluminum parabolic reflector 1/16 inch thick and 16



Sectional Drawing of Complete Flood Light Projector

inches in diameter, mounted on an iron frame. The front of the reflector is covered with a piece of round heat-resisting glass which is clamped to the reflector frame in such a manner that the whole is waterproof. A 500-watt, focus-type, nitrogen-filled lamp is employed with the projector. The spread of the beam can be controlled by moving the lamp in and out of focus and the beam may be concentrated to about 6 degrees divergence with an apparent candle power in the center of slightly over 400,000 by placing the filament exactly at the focus, or by drawing the filament behind the focus, the beam may be spread to 18 degrees with an apparent candle power of approximately 150,000 in the center of the beam. The entire outfit weighs about 30 pounds.

PERSONAL MENTION

Operating Engineers

Melvin Leap, of Cincinnati, Ohio, has been assigned to the river stamer *J. M. Ware* as chief engineer, vice Nora Ellis, retired.

Samuel Ellis, of Cincinnati, Ohio, has been appointed first assistant engineer on the river steamer *Mayme Barrett,* at Cairo, Ill.

C. S. Follett has been elected president of the Marine Engineers' Beneficial Association No. 38, Seattle, Wash., succeeding J. S. Wright. The other officers of this branch of the association for the coming year are F. A. Farr, first vice-president; George W. Hill, second vice- president; W. B. Jackling, corresponding secretary; W. M. Coombs, financial secretary and treasurer.

Changes in United States Steamboat Inspection Service

Chester W. Willett has been appointed assistant inspector of hulls at Milwaukee, Wis., and has been assigned to Chicago for duty.

Thomas E. Kell, local inspector of boilers at St. Michael, Alaska, was killed recently in an automobile accident.

John W. Fleming, assistant inspector of boilers of New York, died recently.

Naval Architects, Consulting Engineers, Draftsmen, and Shipyard and Steamship Officials

Maynard D. Church has been appointed chief engineer of the Terry Steam Turbine Company, Hartford, Conn.

Henderson Weir, secretary of the Harlan & Hollingsworth Corporation, Wilmington, Del., has been elected a director of the corporation.

S. K. Smith has been elected vice-president of the Harlan & Hollingsworth Corporation, Wilmington, Del., succeeding Persifor Frazer. Mr. Smith is also treasurer of this corporation.

George A. Swain, who has been connected with the Southern Pacific Company in New York for nearly thirty years, has been appointed terminal superintendent of the Southern Pacific piers at New York. John Mackin has been appointed assistant to Mr. Swain.

W. J. A. London, for many years connected with the steam turbine business, has become works manager for the Sterling Blower Company, mechanical experts in the manufacture and installation of blowers and blower systems of all kinds, at Hartford, Conn.

E. C. Waldvogel, who has been connected with the Yale & Towne Manufacturing Company, New York, for the past eleven years as traveling salesman, sales manager and assistant general manager, has been appointed general manager of the company, with general supervision of all sales, domestic, Canadian and export.

OBITUARY

Wilfred R. Dean, vice-president of the Dean Bros.' Pump Works, Indianapolis, Ind., died recently, after a protracted illness, aged sixty-eight years.

George D. Rollins, for several years general manager of the Holmes Metallic Packing Company, Wilkesbarre, Pa., died in Philadelphia on December 16, aged eighty. Mr. Rollins became associated with the Holmes Metallic Packing Company in 1901, and served as general manager until the time of his death.

Colonel Edwin D. Metcalf, president and founder of the Columbian Rope Company, Auburn, N. Y., died at his home in that city on December 31, aged sixty-seven. Colonel Metcalf came to Auburn from New England in 1890 to become treasurer and general manager of the D. M. Osborne Company, manufacturers of farm machinery. In 1903 the International Harvester Company bought the plant of the Osborne Company, and Mr. Metcalf established the Columbian Rope Company, which he developed until to-day it ranks among the largest and best-known cordage factories in the world.

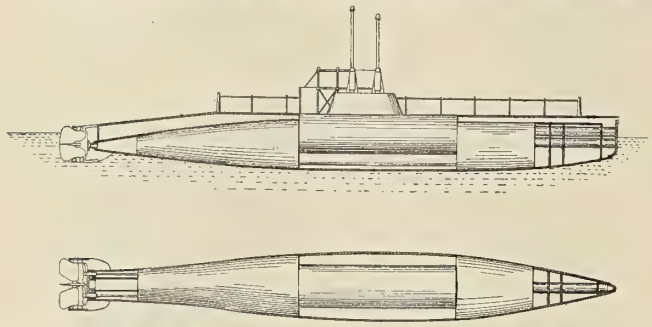
SELECTED MARINE PATENTS

The publication in this column of a patent specification does not necessarily imply editorial commendation.

American patents, compiled by Delbert H. Decker, Esq., registered patent attorney, Millerton, N. Y.

1,154,215. HULL CONSTRUCTION OF SUBMARINE BOATS. LAWRENCE Y. SPEAR, OF QUINCY, MASS., ASSIGNOR TO ELECTRIC BOAT COMPANY, OF NEW YORK, N. Y., A CORPORATION OF NEW JERSEY.

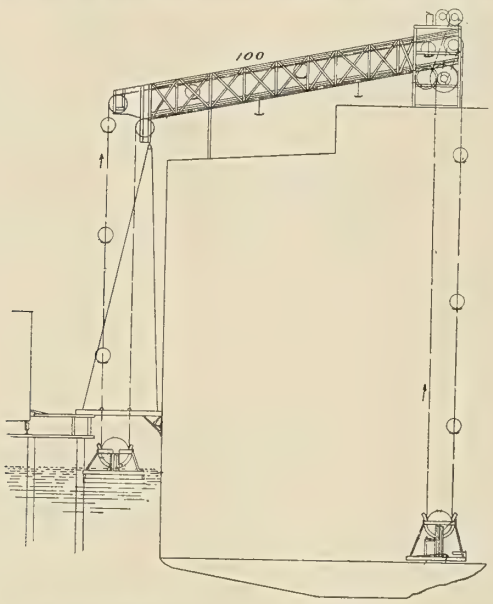
Claim 1.—A hull construction for submarine boats, comprising a double hulled midship section consisting of an inner strong hull and an outer weak hull, and a single strong hull section forward and one aft of said midship section, the center line of the volume of the inner



strong hull of the midship section lying above the center line of the volume of the forward and aft strong hull sections, said single strong hull sections forming continuations of the weak hull of said midship section. 18 claims.

1,156,653. SHIP LOADER OR UNLOADER. BENJAMIN H. ALVEY, OF ELIZABETHTOWN, KY.

Claim 3.—A ship loader or unloader comprising an endless traveling conveyor guided to provide an intermediate section and sections depending from the ends thereof, said conveyor having pivotally suspended package carriers, a loading device supported in operative relation with one of said end sections, said loading device being provided with means



for supporting packages in the path of the package carriers and arranged to be moved on its support with swinging movement of said section, the latter having traveling movement relatively to said device and extending below said device and being freely suspended. Thirty-one claims.

1,165,183. DAVIT-FALL ATTACHMENT. FERDINAND W. KROGH, OF SAN FRANCISCO, CAL.

Claim 1.—A davit-fall attachment comprising companion hooks engaged together and means between said hooks to release said engagement, one of said hooks being secured to the fall and the other to the boat. Eight claims.

1,153,475. MARINE TURBINE. KARL ALQUIST, OF SCHENECTADY, N. Y., ASSIGNOR TO GENERAL ELECTRIC COMPANY, A CORPORATION OF NEW YORK.

Claim 1.—In a sytem of ship propulsion, the combination of main turbines arranged to propel the ship at maximum speed, cruising turbines arranged to co-operate with the main turbines to propel the ship at speeds below the maximum, and means arranged to connect the cruising turbines in series with the main turbines as regards the flow of steam for intermediate speed conditions and to connect the cruising turbines serially as regards the flow of steam and to supply the exhaust therefrom to the main turbines in parallel for lower speeds of the ship. 7 claims.

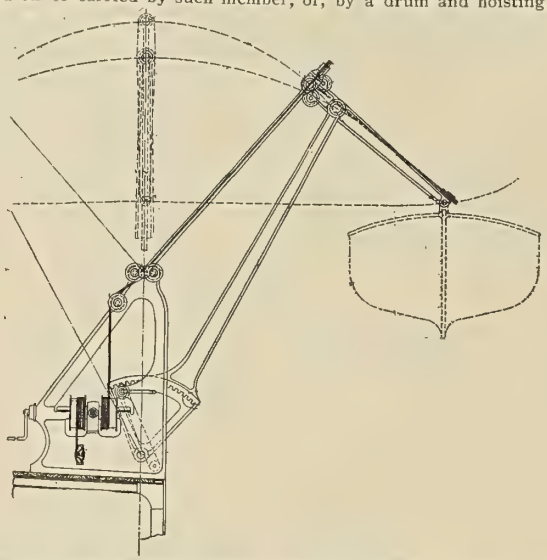
British patents compiled by G. F. Redfern & Co., chartered patent agents and engineers, 15 South street, Finsbury, E. C., and 10 Gray's Inn place, W. C., London.

5674/14. APPLIANCE FOR RENDERING SHIPS UNSINKABLE. A. PIERROTTET, 31 VIA XX SETTEMBRE, GENOA, ITALY, ENGINEER.

According to the invention the unserviceable spaces such as the spaces between the framings, the cargo battens, the bulkhead stiffeners, the deck beams, and so on, are lined, or if desired are wholly or partly filled with a compound of small specific weight, such as cork chips, sawdust and the like, kneaded up with tar, cement, or any like substance, the compound when desired being made lighter by hollow spaces inside the mass, which is applied either in the form of a paste or as a concrete. The volume, weight and arrangement of the whole is such that on one or several compartments being flooded, the material will suffice to insure the buoyancy and the stability of the ship, either until those on board have been rescued or under certain circumstances until the ship itself has reached port.

20,813/14. SHIPS' DAVITS. J. M. DESCHAMPS, OF 63, CHANCERY LANE, LONDON, MIDDLESEX.

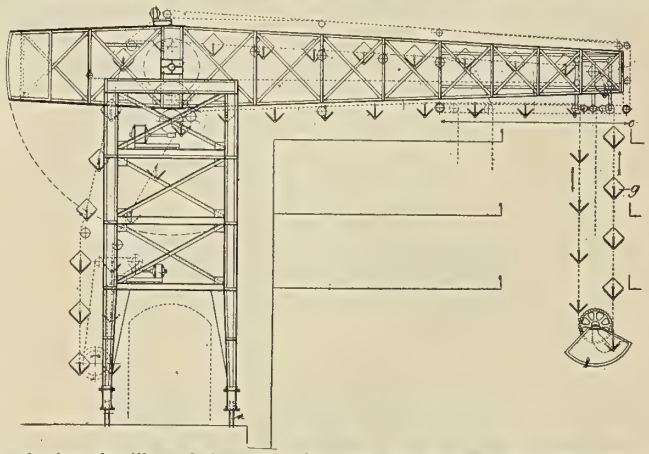
This invention relates to davits for ships' boats of the type in which the main supporting member is actuated; that is, thrown inboard or outboard past the fixed frame or support, by means of a toothed rack formed on or carried by such member, or, by a drum and hoisting chain



or rope, the load carrying member being controlled by the movement of said main supporting member and so mounted thereon that the free end of the load carrying member travels in an approximately straight line. The objects are to enable an enlarged throw inboard and outboard to be obtained without excessive height, to maintain a straight path of travel for the boat to enable several tiers of boats to be handled and to combine, if desired, with the davits a lowering device for operating the falls and the load, preferably with single wire or other ropes.

1,145/15. LOADING AND DISCHARGING THE SHIPS' CARGOES. A. FRANKLIN, OF 70, WILMINGTON GARDENS, NEW BARKING, ESSEX.

A crane jib or a balanced girder is mounted upon a traveling base in combination with a conveyor or elevator running along the jib or girder and depending from each end thereof, the arrangement being preferably



such that the jib or balanced girder with the elevator depending therefrom may be moved from a more or less vertical position to an inclined or horizontal position in which the outer end of the jib or balanced girder may be adjusted as to height and thereby adjust the pending end of the elevator to the required height in the ship's hold to receive or deliver goods to be handled. The said combined jib or girder and conveyor is preferably arranged in relation to another conveyor delivering or receiving goods from the adjacent sheds or warehouses. If required, the elevator itself may be arranged to be adjustable as to length independently of the position of the crane jib or balanced girder.